PB-213 389

AUTOMOBILE GAS TURBINE OPTIMIZATION STUDY

AiResearch Manufacturing Company of Arizona
Phoenix, Arizona

Prepared for:

Environmental Protection Agency

14 July 1972

DISTRIBUTED BY:

NTIS
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U. S. DEPARTMENT OF COMMERCE
5285 Port Royal Road, Springfield Va. 22151

APTD-1291

PB 213 389

FINAL REPORT

AUTOMOBILE GAS TURBINE OPTIMIZATION STUDY

Prepared by

AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

(A Division of The Garrett Corporation)
402 S. 36th Street
Phoenix, Arizona 85034

Prepared for the

ENVIRONMENTAL PROTECTION AGENCY
OFFICE OF AIR PROGRAMS

Contract 68-04-0012

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INFORMATION SERVICE
U S Department of Commerce
Springfield VA 22151

AT-6100-R7 July 14, 1972

R
414

BIBLIOGRAPHIC DATA SHEET

1. Report No. APTD-1291

2.

3. Recipient's Accession No. PB-213 384

4. Title and Subtitle
Automobile Gas Turbine Optimization Study

5. Report Date
July 14, 1972

6.

7. Author(s)

8. Performing Organization Rept. No. AT-6100-R7

9. Performing Organization Name and Address
AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A Division of The Garrett Corporation
402 S. 36th Street
Phoenix, Arizona 85034

10. Project/Task/Work Unit No.

11. Contract/Grant No.
68-04-0012

12. Sponsoring Organization Name and Address
ENVIRONMENTAL PROTECTION AGENCY
Office of Air Programs
Research Triangle Park, North Carolina 27711

13. Type of Report & Period Covered
Final

14.

15. Supplementary Notes

16. Abstracts

A study was conducted for the purpose of defining the optimum gas turbine engine capable of meeting the 1976 emission standards, when used in the largest-selling size classification of U.S. automobiles and capable of being developed by the year 1975. This study approach was to perform limited analyses, consisting of design-point cycle studies of a large number of potential candidate automobile engines comprising single-shaft, free-turbine and multiple spool configurations. Cycle pressure ratios and temperatures, with turbine cooling flow rates, were varied, thus providing sufficient information to estimate qualitatively the part load performance of each engine. In these analyses, the effects of cycle modifiers were determined. Twelve cycles were chosen for off-design performance analysis over the entire operating range of each engine. These engine performance curves, with performance curves of suitable transmissions, were used in a computer vehicle simulation program to compute vehicle acceleration performance, fuel economy for selected driving schedules, and emission estimates. Engine and transmission design layouts were made, and direct manufacturing costs were estimated for three systems which were finally chosen as prime candidates. Maintenance and repair, fuel, and other costs were estimated for the 7-year, 105,000-mi life of the vehicle, to obtain a vehicle cost-of-ownership.

17. Key Words and Document Analysis. 17a. Descriptors

Air pollution
Gas turbine engines
Automobiles
Emission
Standards
Design
Cycles
Performance evaluation
Operating costs
Automotive transmissions
Economic analysis
Combustion chambers
Compressors
Turbines
Regenerators

17b. Identifiers/Open-Ended Terms

Ownership
Mission analysis
Off-design performance

17c. COSATI Field/Group 13B

18. Availability Statement
Unlimited

19. Security Class (This Report)
UNCLASSIFIED

20. Security Class (This Page)
UNCLASSIFIED

21. No. of Pages
425

22. Price

AUTOMOBILE GAS TURBINE OPTIMIZATION STUDY
FOR THE
ENVIRONMENTAL PROTECTION AGENCY

AT-6100-R7 July 10, 1972

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GARRETT
AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION

REPORT NO. AT-6100-R7

TOTAL PAGES 420

ATTACHMENTS:

REV	BY	APPROVED	DATE	PAGES AND/OR PARAGRAPHS AFFECTED

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SECTION 1

INTRODUCTION AND SUMMARY

1.1 INTRODUCTION

The effort reported herein was performed by the AiResearch Manufacturing Company of Arizona, A Division of The Garrett Corporation, on the Automobile Gas Turbine Optimum Configuration Selection Study under Contract 68-04-0012 for the Environmental Protection Agency, Office of Air Programs, Advanced Automotive Power Systems Development Division.

The purpose of this study was to define the optimum gas turbine engine capable of meeting the 1976 emission standards, when used in the largest-selling-size classification of U.S. automobiles, and capable of being developed by the year 1975. Optimization criteria were:

(a) Emissions - 1976 Federal Standards as specified in the July 2, 1971, Federal Register

(b) Performance - Similar to present automobiles as specified in Appendix 1

(c) Cost of Ownership - Equal to or less than 110 percent of present standard-size automobile with spark-ignition engine as determined by the U.S. Department of Commerce 1969/70 statistics

(d) Mean Operational Life - 7 years, 105,000 mi, 3500 hr

(e) Installation Volume - 35 ft^3 maximum

1.2 SUMMARY

This study approach was to perform limited analyses, consisting of design-point cycle studies of a large number of potential candidate automobile engines comprising single-shaft, free-turbine, and multiple-spool configurations. Cycle pressure ratios and temperatures, with turbine cooling flow rates, were varied, thus providing sufficient information to estimate qualitatively the part-load performance of each engine. In these analyses, the effects of cycle modifiers, such as heat exchangers and variable compressor and turbine geometry, were determined. Over 150 cycles were analyzed in the above process.

Twelve of the promising cycles were chosen for off-design performance analysis over the entire operating range of each engine. These performance curves for each engine, with performance curves of suitable transmissions, were used in a computer vehicle simulation program to compute vehicle acceleration performance, fuel economy for selected driving schedules, and emissions estimates.

Three of these engine-transmission systems were then chosen as prime candidates. Engine and transmission design layouts were made, and direct manufacturing costs of each system were estimated. Maintenance and repair, fuel, and other costs were estimated for the 7-year, 105,000-mi life of the vehicle, to obtain a vehicle cost-of-ownership.

Results of this study show that a relatively low pressure ratio, single-shaft engine with regeneration (either a fixed-boundary recuperator or a rotary regenerator, driving the vehicle through an infinitely variable speed-ratio traction transmission) is the system most capable of meeting or exceeding the optimization criteria. To minimize fuel consumption and, therefore, total cost of ownership, a high heat recovery effectiveness is recommended--85 percent for a fixed-boundary recuperator and 90 percent for a rotary regenerator.

Although consumer initial cost would be decreased by using a lower heat recovery effectiveness, the higher values are required to minimize the cost-of-ownership optimizing parameter.

Additional work, beyond the scope of this study, is required to determine the optimum heat recovery device. The three most promising candidates for the heat exchanger are the ceramic rotary regenerator, the ceramic counterflow recuperator, and the metallic counterflow recuperator. The primary problem with the rotary regenerator is that of designing the gas path seals to minimize the leakage loss to the cycles and maintain an adequate seal life at an acceptable cost. This problem has been investigated for several years, with evidence of apparent solution. However, the proprietary nature of this work affords little specific information to support this contention. Because of the high heat recovery effectiveness ($\epsilon_R \leq 0.90$) obtainable at a reasonable weight and volume and potentially low cost, the rotary regenerator is the preferred heat exchanger for the standard automobile-type gas turbine engine; assuming, of course, that the seal problem is resolved.

However, if the regenerator seal problem has not been adequately solved (as contended by some sources) and assuming the solution is not imminent, some other type of heat exchanger must be used. For an effectiveness of 0.85, the counterflow recuperator is the preferable unit. Two types of recuperator core materials were considered in the analysis, ceramic and metallic. The ceramic material offers a potential for lower initial cost but has not been proven as a practical material for the counterflow configuration. For the metallic recuperator, 347 Stainless Steel was used for the core material; as a result, the estimated initial cost is relatively high. However, a concept has been proposed to reduce this high cost. If this concept is proven practical, the metallic recuperator could be more attractive than ceramic versions.

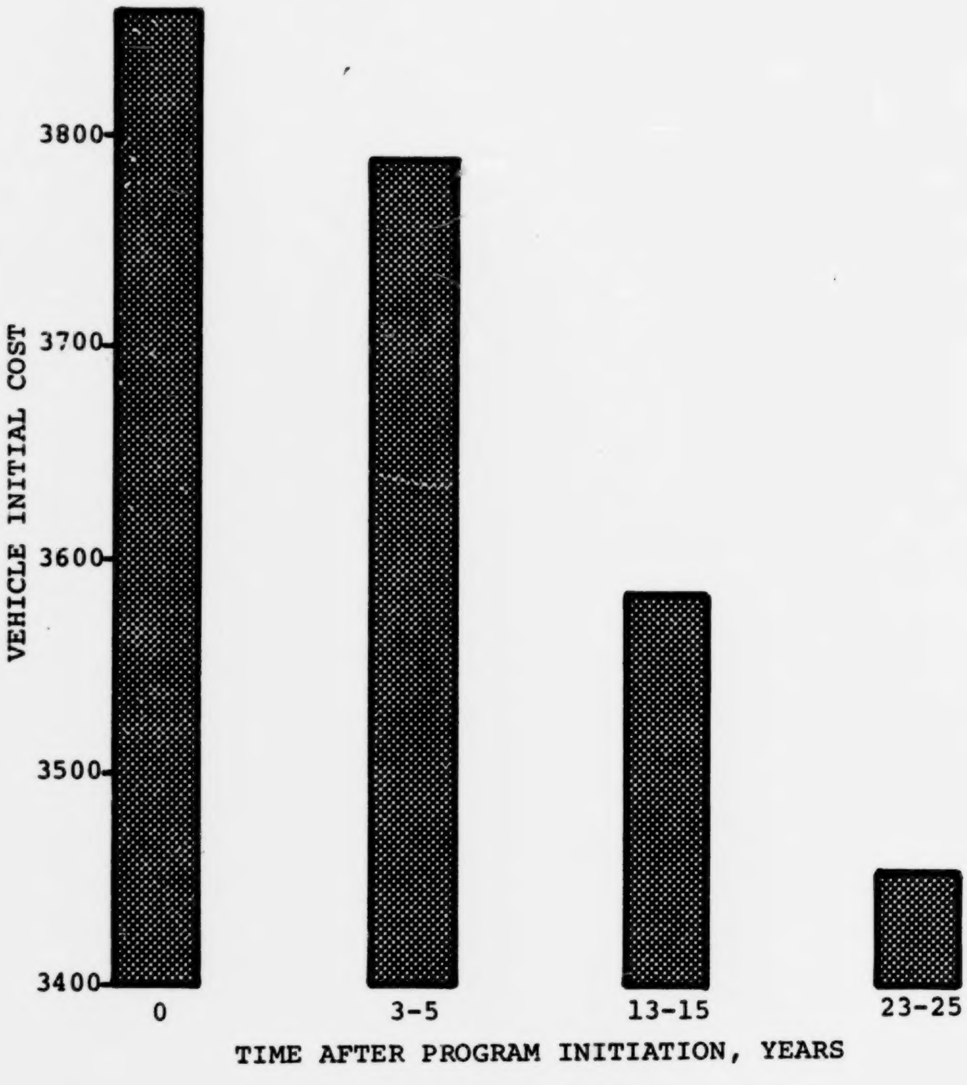
Further development of the three heat exchangers will identify the optimum heat recovery device. The recommended engine configuration does not depend on which of the three heat exchangers is selected. In each case, the recommended engine is of a single-shaft configuration; only the optimum design-point pressure ratio would be slightly affected by the heat recovery effectiveness chosen.

The 7-year, 105,000-mi estimated cost-of-ownership of the automobile with the selected gas turbine engine is less than that for a corresponding 1970 automobile with spark-ignition engine. Based on operating life data for aircraft gas turbine engines, the mean life of the automobile gas turbine will exceed the 3500-hr, 105,000-mi specification.

The optimum gas turbine engine is easily packaged in the limiting 35-ft^3 volume. An installation drawing showing one version of the engine-transmission system installed in a standard-size U.S. automobile is included in Section 10.

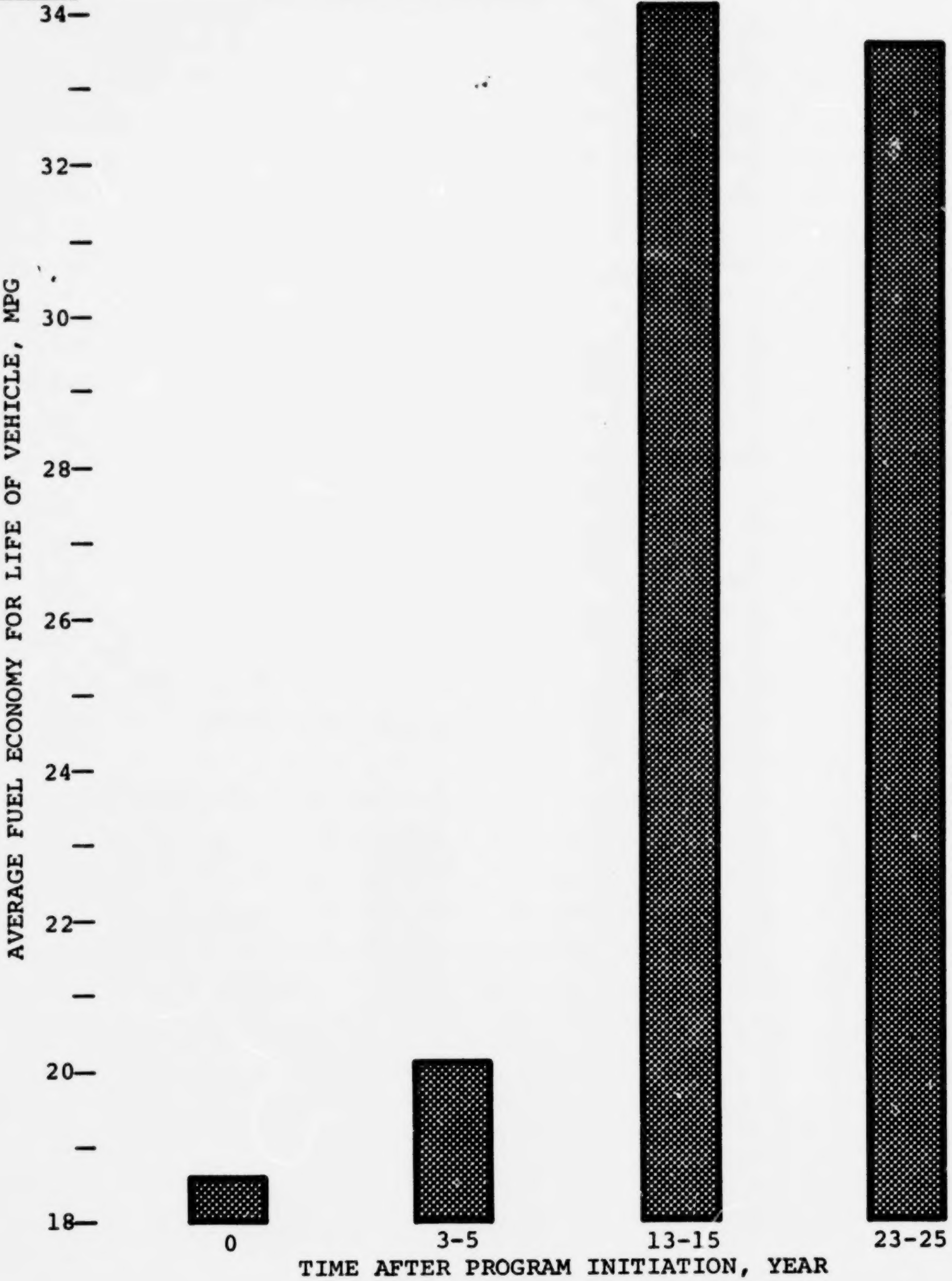
The development program required to achieve the projected 1975 technology level is provided in Section 11.

The long range potential of the gas turbine engine automobile is analyzed. Figures 1-1 through 1-3 summarize the results of this analysis. The vehicle initial cost, in terms of 1970 dollars, is estimated to decline as new materials are developed for use in the engine and as component efficiencies are improved, decreasing the size of the engine. Major reductions in cost will result from substituting ceramic materials for high-temperature nickel and cobalt base alloys. The development of a ceramic turbine will allow higher turbine inlet temperatures, resulting in better efficiency and lower fuel costs.



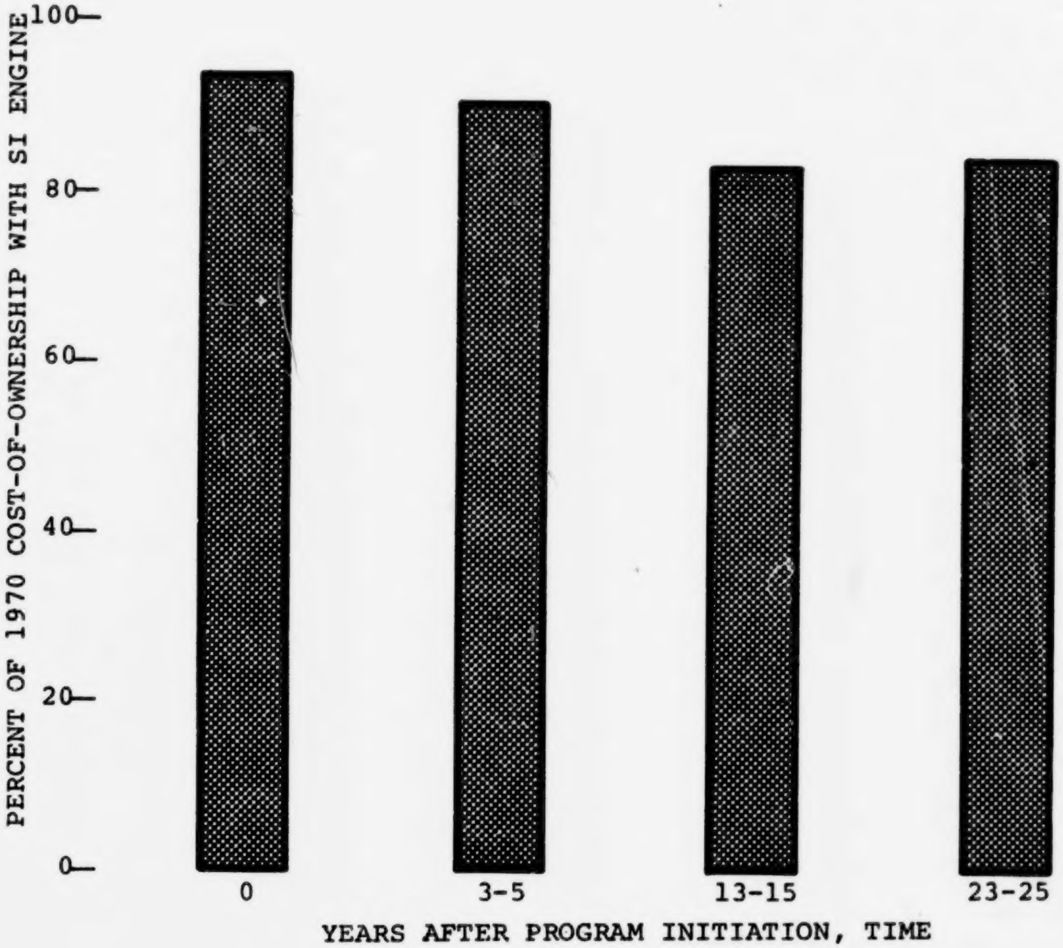
VEHICLE INITIAL COST TREND FOR OPTIMUM
GAS TURBINE AUTOMOBILE

FIGURE 1-1



COMPOSITE ROUTE FUEL ECONOMY TREND FOR OPTIMUM GAS TURBINE AUTOMOBILE

FIGURE 1-2



COST-OF-OWNERSHIP TREND FOR OPTIMUM GAS TURBINE AUTOMOBILE

FIGURE 1-3

The average fuel economy of the gas turbine automobile as projected will increase with increasing component efficiencies and turbine inlet temperatures (Figure 1-2). Increasing fuel economy is a significant factor, since a fuel crisis is anticipated in the next 20 to 30 years.

The cost of ownership, in terms of 1970 dollar-value, is expected to slightly decrease with the projected lower initial cost of the engine and improved fuel economy (Figure 1-3).

This is a limiting case, in that the engine cycles were not penalized for low-emission combustor designs, simply because this penalty was not known during the course of the study. Results of the low-emission combustor programs, conducted in parallel with this program, will provide the performance penalties, if any, for the low-emission combustor designs.

Federal Driving Cycle (FDC) emissions for the three candidate systems were computed, using data from AiResearch aircraft engines and employing analytical techniques. All of the engines met the CO and UHC requirements with margin, but none of these conventional combustors met the 1976 NO_x requirement. The recuperated engine had the lowest NO_x emission, as expected, because of the lower combustor inlet temperature.

SECTION 2

BASELINE TECHNOLOGY

Vehicle performance requirements specified in Paragraph 8 of "Vehicle Design Goals - Six-Passenger Automobile," reprinted in Appendix 1, were used to establish power requirements of the engine at approximately 150 hp or an airflow rate of 1 to 2 lb/sec. This requirement was then used to calculate the approximate engine component size and design-point performance characteristics, as discussed.

2.1 COMPRESSORS

The following candidate compressor configurations were considered in the optimum cycle studies:

(a) Single-state centrifugal

(b) Single-stage mixed-flow

(c) Axial first-stage, centrifugal second-stage

(d) Axial first-stage, mixed-flow second-stage

(e) Centrifugal first-stage, centrifugal second-stage

The impeller, diffuser, and stage-matching of all candidate compressors was optimized to give the best performance possible at low part-load power, since the average speed of the automobile for its operating-life is approximately 30 mph when the engine output power is less than 20 percent of the peak power.

2.1.1 Current Technology

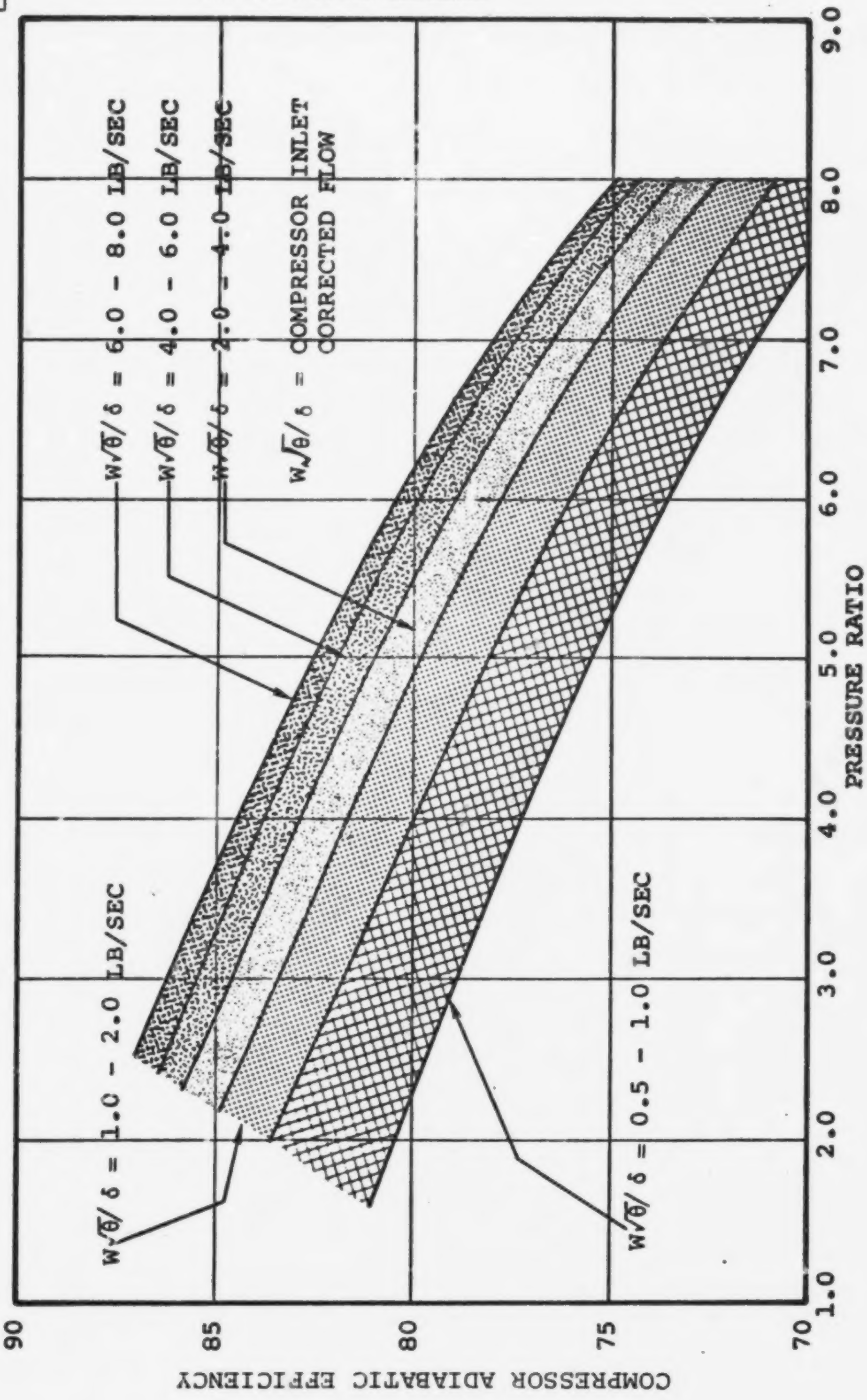
The interrelationship of pressure ratio, flow rate, and adiabatic efficiency of various AiResearch production centrifugal compressors is shown in Figure 2-1. The curves were obtained from test data of compressors with backward-curved impeller blades and low compressor exit Mach number. This figure shows the quantitative effects on performance associated with increased pressure ratio, as well as the size effects (tolerances and running clearances), which influence both the impeller and diffuser.

High Mach numbers, relative to the inducer tip and diffuser inlet, are associated with high-pressure ratio centrifugal compressors. As a result, backward-curved blades are used for a wide surge margin at high inducer tip relative Mach numbers. The impeller exit absolute Mach number is also lower with backward-curved impeller blades than with radial blades.

Figure 2-2 shows the current state of the art of various single-stage axial compressors. These data include stage efficiency, pressure ratio, surge margin, tip speed, and flow rates for the design and some off-design speeds.

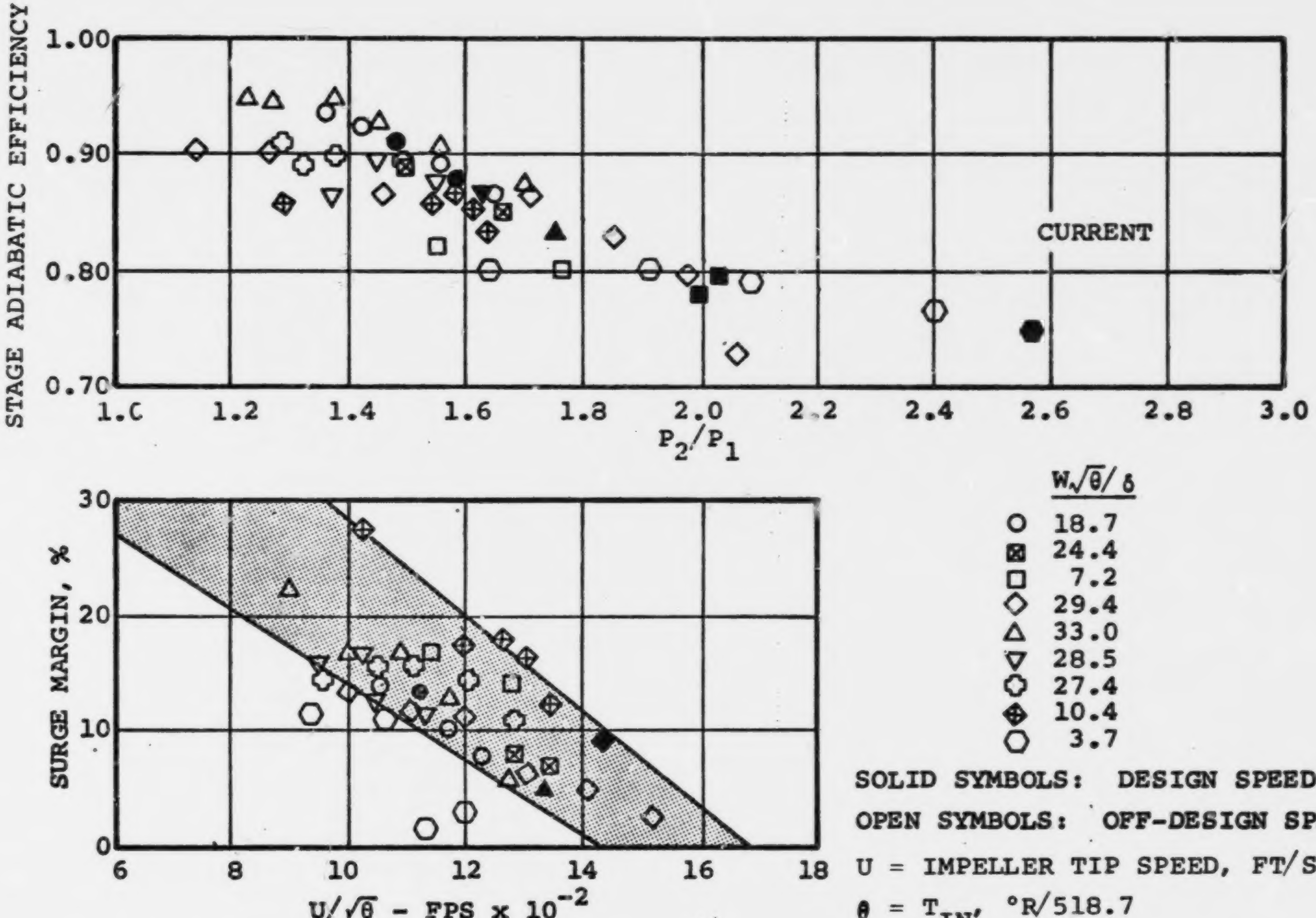
The mixed-flow compressor is a form of the centrifugal compressor and differs only in that the flow exits from the impeller less than 90 degrees from the axis of rotation. AiResearch accomplished an investigation of mixed-flow compressor design criteria and ascertained that mixed-flow and radial out-flow compressors can be designed using identical techniques and will yield comparable performance for the same design point.

Figures 2-3 and 2-4 show compressor adiabatic efficiencies, as a function of pressure ratio, that were used in the design-point cycle studies for one- and two-stage compressor configurations.



SINGLE-STAGE CENTRIFUGAL COMPRESSOR

FIGURE 2-1



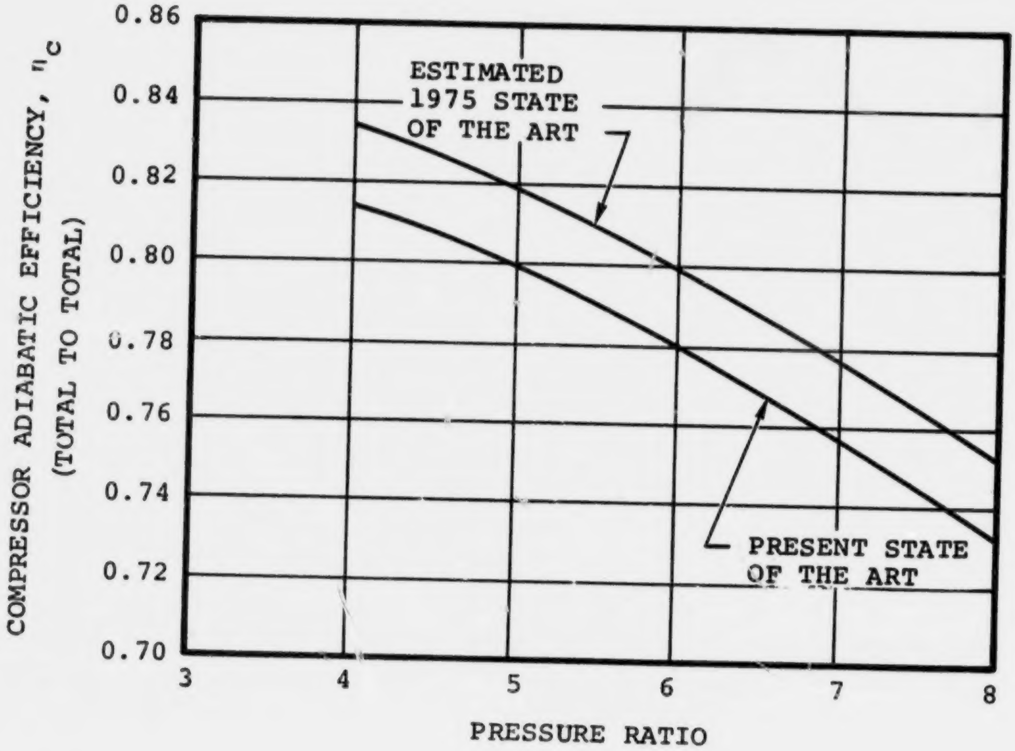
SOLID SYMBOLS: DESIGN SPEED DATA

OPEN SYMBOLS: OFF-DESIGN SPEED DATA

U = IMPELLER TIP SPEED, FT/SEC
θ = T_{IN}, °R/518.7
S = P_{IN}, PSIA/14.696

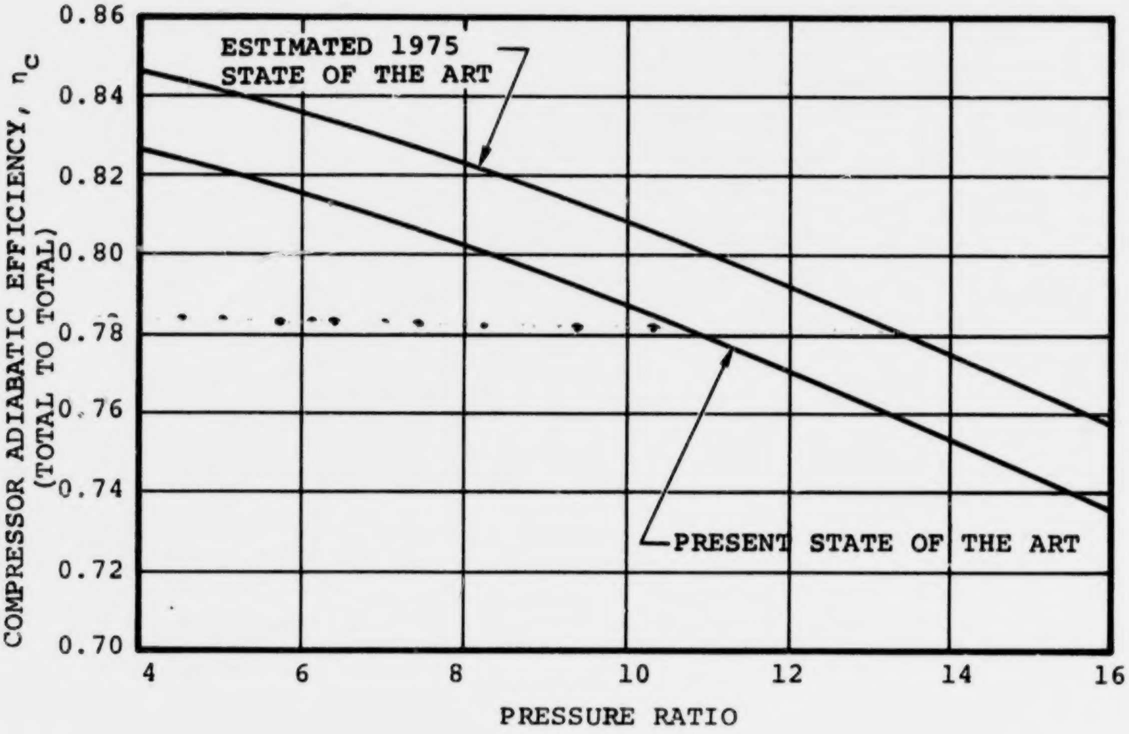
SINGLE-STAGE AXIAL COMPRESSOR EFFICIENCY ESTIMATES

FIGURE 2-2



COMPRESSOR PERFORMANCE, SINGLE-STAGE RADIAL OR MIXED FLOW

FIGURE 2-3



COMPRESSOR PERFORMANCE, TWO STAGES;
AXIAL-RADIAL OR TWO-RADIAL

FIGURE 2-4

2.1.2 1975 Established Technology

The 1975 compressor performance projection is a two-point improvement in efficiency. This improvement is expected to be made by better contour control, improved design technology, and improved control of manufacturing tolerances. These improvements will be applicable to mass production components, as well as small quantity production.

2.2 COMBUSTORS

Federal emission standards specify the following levels of exhaust pollutants for 1976 automobiles:

(a)	Hydrocarbons	- 0.41 gm/mi
(b)	Carbon monoxide	- 3.4 gm/mi
(c)	Oxides of nitrogen	- 0.4 gm/mi

These pollutants are the result of incomplete combustion, particularly the hydrocarbons (HC) and carbon monoxide (CO). Therefore, pollutant levels can be utilized to compute the combustion efficiency required for the Federal Driving Cycle to meet the specified limits.

To calculate combustion efficiency, levels in grams per mile must be converted to an equivalent mission index expressed as milligrams of pollutant per gram of fuel. The formula for accomplishing this conversion is

$$EI = 1000 \times \frac{gal}{cm^3} \times \frac{1}{gm/cm^3} \times \frac{mi}{gal} \times \frac{gm}{mi}$$

where:

EI = emission index, mg/gm

$\text{gal/cm}^3 = 1/3785$

gm/cm^3 = density of fuel = 0.763 gm/cm^3

Combining the constant values yields

$$EI = 0.34626 \times \frac{\text{mi}}{\text{gal}} \times \frac{\text{gm}}{\text{mi}}$$

For example, if average fuel consumption over the Federal Driving Cycle (FDC) is 10 mpg, the emission index for each pollutant is

	FDC gm/mi	EI
HC	0.41	1.42
CO	3.4	11.80
NO_x	0.4	1.38

Combustion efficiency is variously expressed as (1) actual combustor temperature rise divided by ideal temperature rise for a given fuel-air ratio, (2) ideal fuel-air ratio divided by actual fuel-air ratio for a given temperature rise, or (3) actual enthalpy rise divided by ideal enthalpy rise. There is very little difference in these three values unless efficiency is less than 90 percent. For the purpose of emission calculations, efficiency must be on an enthalpy basis. For an adiabatic combustion reaction, the sensible enthalpy change is equal to the chemical energy released in reaction. Therefore, the combustion efficiency can be expressed as the chemical enthalpy change for the actual products divided by that of the ideal.

The ideal chemical enthalpy change on a pound-of-fuel basis is equal to the fuel lower heating value. The same actual change is the lower heating value of the fuel minus that of each standard product formed on a pound-of-fuel basis. The combustion efficiency can then be written as

$$\eta = \frac{LHV_f - \sum M_p HV_p}{LHV_f} \times 100$$

where:

η = combustion efficiency, percent

LHV_f = fuel lower heating value, Btu/lb

M_p = pounds of pollutant per pound of fuel

HV_p = heating value of pollutant, Btu/lb

For JP-4 fuel, LHV_f = 18,500 Btu/lb; for carbon monoxide, HV_p = 4345.23 Btu/lb; for hydrocarbon as $CH_{1.85}$ per the Federal Test requirements, HV_p = 18,656 Btu/lb. The influence of oxides of nitrogen is so small that it can be disregarded for purposes of calculating required combustion efficiency. Therefore,

$$\eta = \frac{18{,}500 - (4345.23 \times M_{CO}) - (18{,}656 \times M_{HC})}{18{,}500} \times 100$$

$$\eta = 100 - (23.48 \times M_{CO}) - (100.8 \times M_{HC})$$

or for emission index,

$$EI = M_{CO} \times 1000$$

$$\eta = 100 - 0.02348 \times EI_{CO} - 0.1008 \times EI_{HC}$$

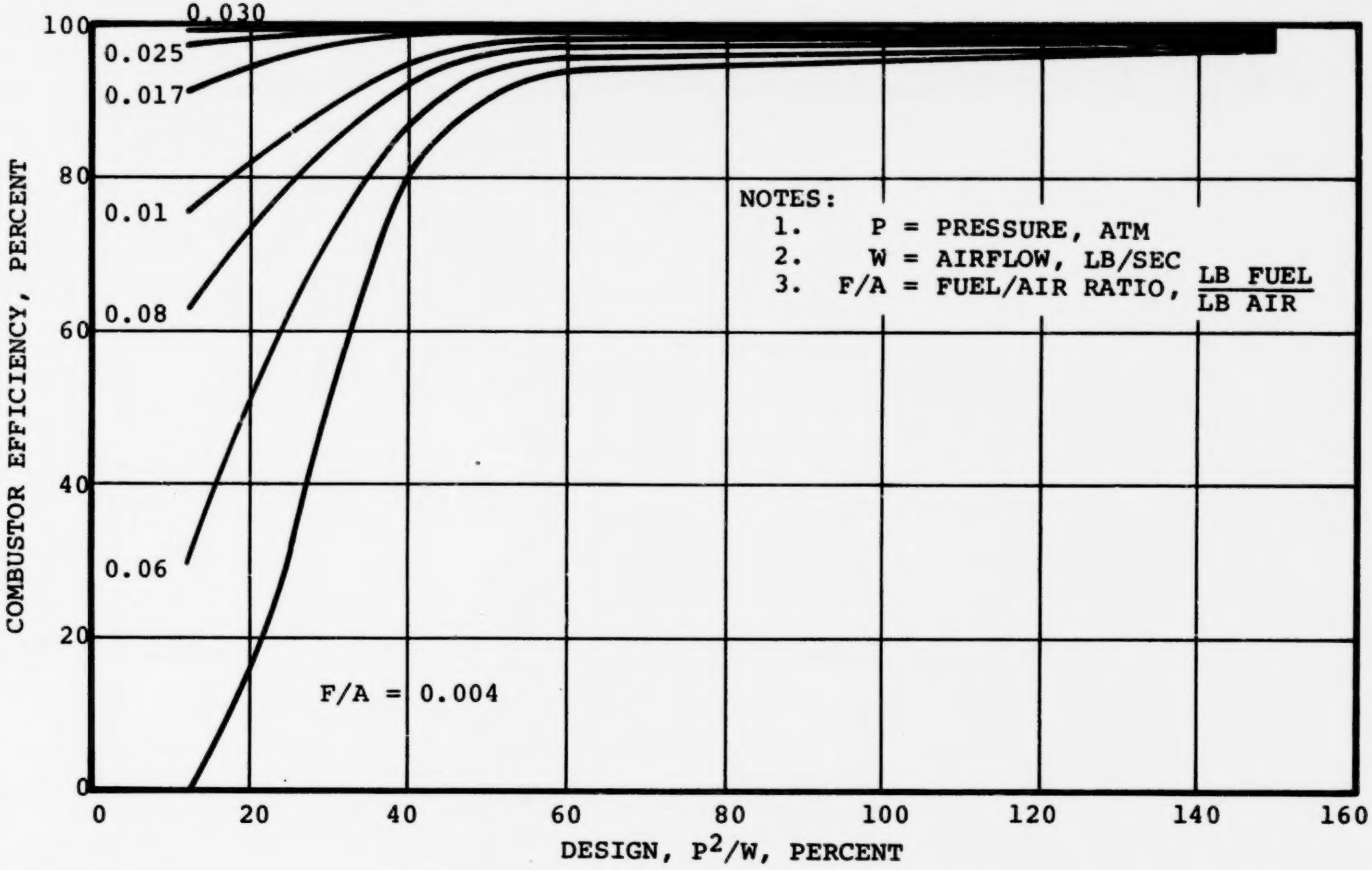
For the specified emission goals and the 10 mpg example

$$\eta = 100 - 0.02348 \times 11.8 - 0.1008 \times 1.42$$

$$\eta = 99.58 \text{ percent}$$

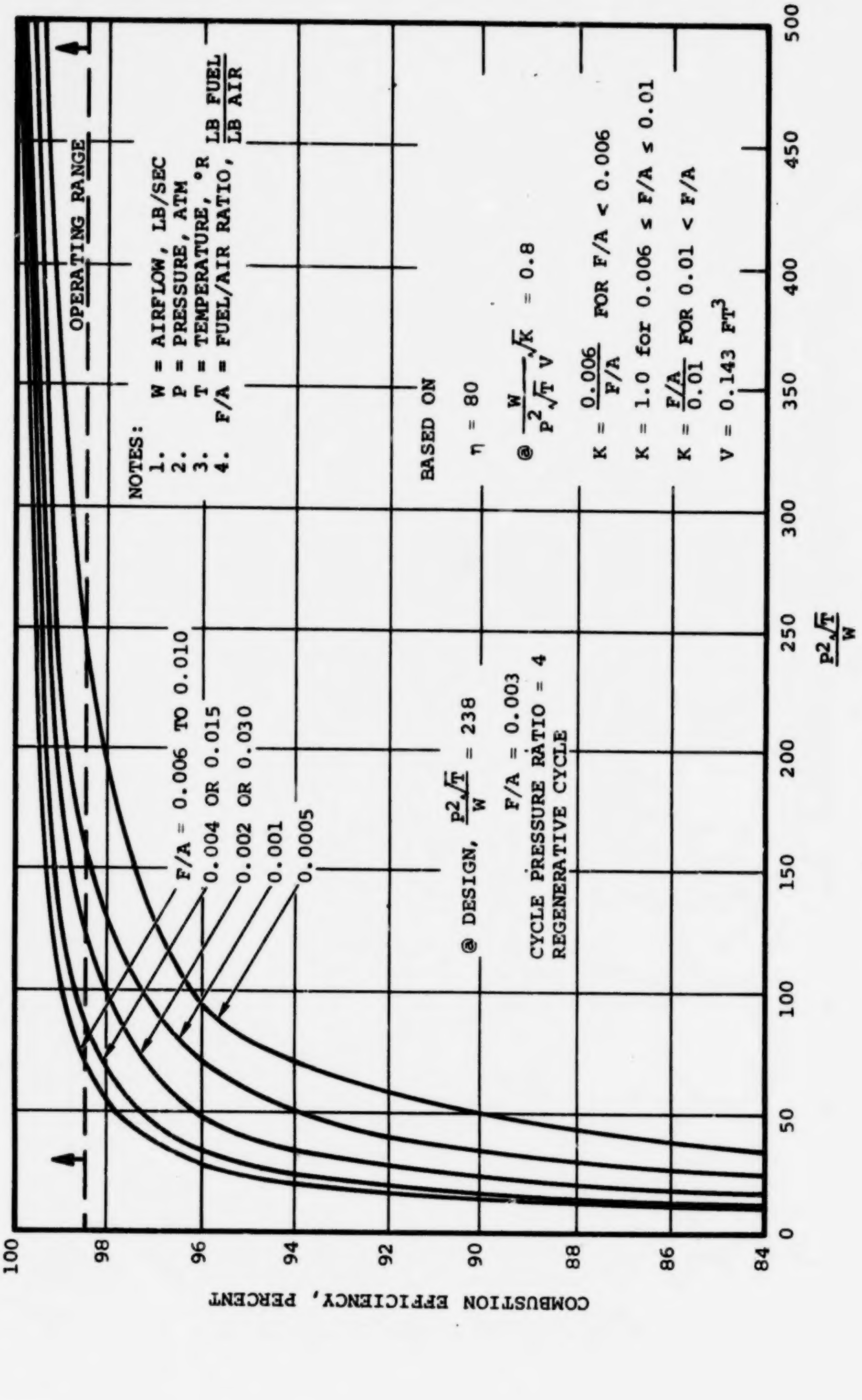
In an actual mission analysis, combustion efficiency and fuel consumption will vary; hence, the preceding represents an overall integrated efficiency for the total amount of fuel burned during the cycle. To achieve this, combustion efficiencies, particularly at idle and part-load, must be on the order of 99.6 percent. This implies that the combustor must be designed for the part-load operating condition rather than for maximum power, to ensure that the required high-efficiency levels be maintained.

With a conventionally sized combustor (i.e., one that is sized at a maximum power-point), efficiency drops rapidly at the off-design low power-points in that higher aerodynamic combustor loading requires increased combustion zone volume. However, the combustor must be sized at a low power-point and will be oversized at maximum power. The combustor map of a conventional aircraft engine, based on experimental data, is shown in Figure 2-5. Maps of low-pressure ratio recuperated and high-pressure ratio unrecuperated engine combustors, designed for high efficiency at part-load, are shown in Figures 2-6 and 2-7, respectively. Note that the loading parameter was modified to include the effect of combustor inlet temperature in the latter maps. By designing combustors for part-load, the volume of those in low pressure must be increased by approximately 90 percent and those in high pressure by approximately 50 percent. The net effect is to place the combustor operating range to the right of the "knee" (as shown on the efficiency curves in Figures 2-6 and 2-7), thereby assuring maximum combustion efficiency over the entire range.



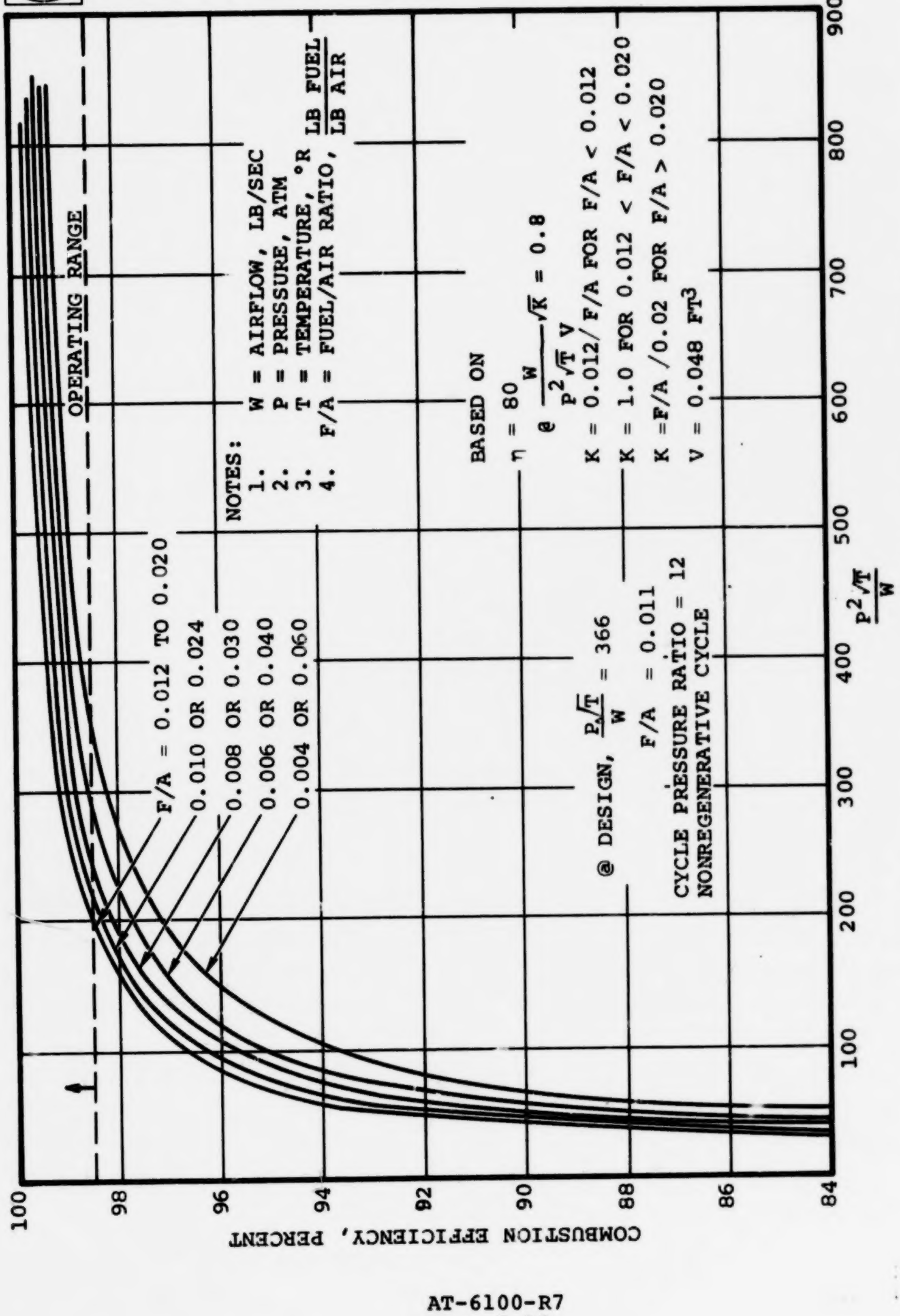
COMBUSTOR EFFICIENCY VS LOADING PARAMETER
(CONVENTIONAL AIRCRAFT ENGINE)

FIGURE 2-5



COMBUSTION EFFICENCY, LOW-PRESSURE RATIO COMBUSTOR

FIGURE 2-6



COMBUSTION EFFICIENCY, HIGH-PRESSURE RATIO COMBUSTOR

FIGURE 2-7

2.3 TURBINES

Candidate turbine configurations considered for free-power turbine engines were:

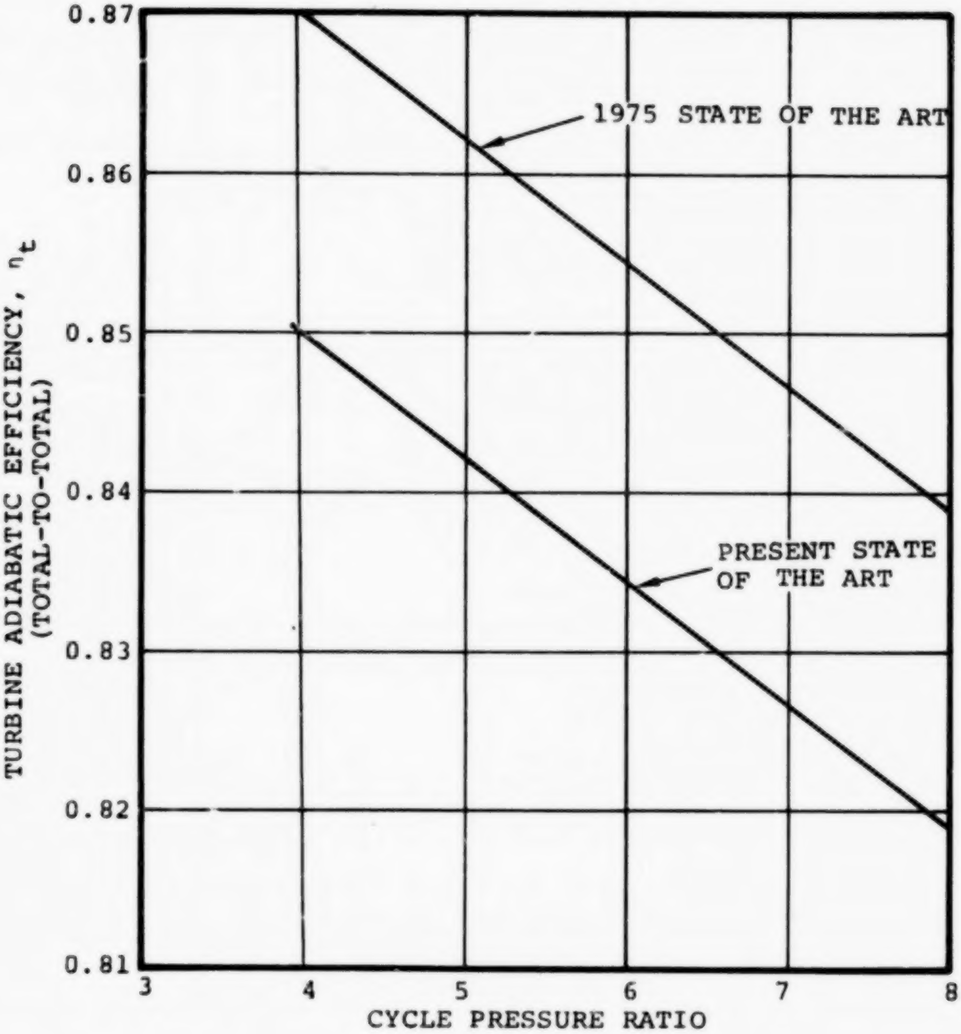
Configuration No.	Gas Generator Turbine	Power Turbine
1	One radial	One axial
2	One axial	One axial
3	Two axial	One axial

Turbines considered for single-shaft engines included the following:

Configuration No.	Turbine
1	One radial
2	One radial and one axial
3	Two axial
4	Three axial

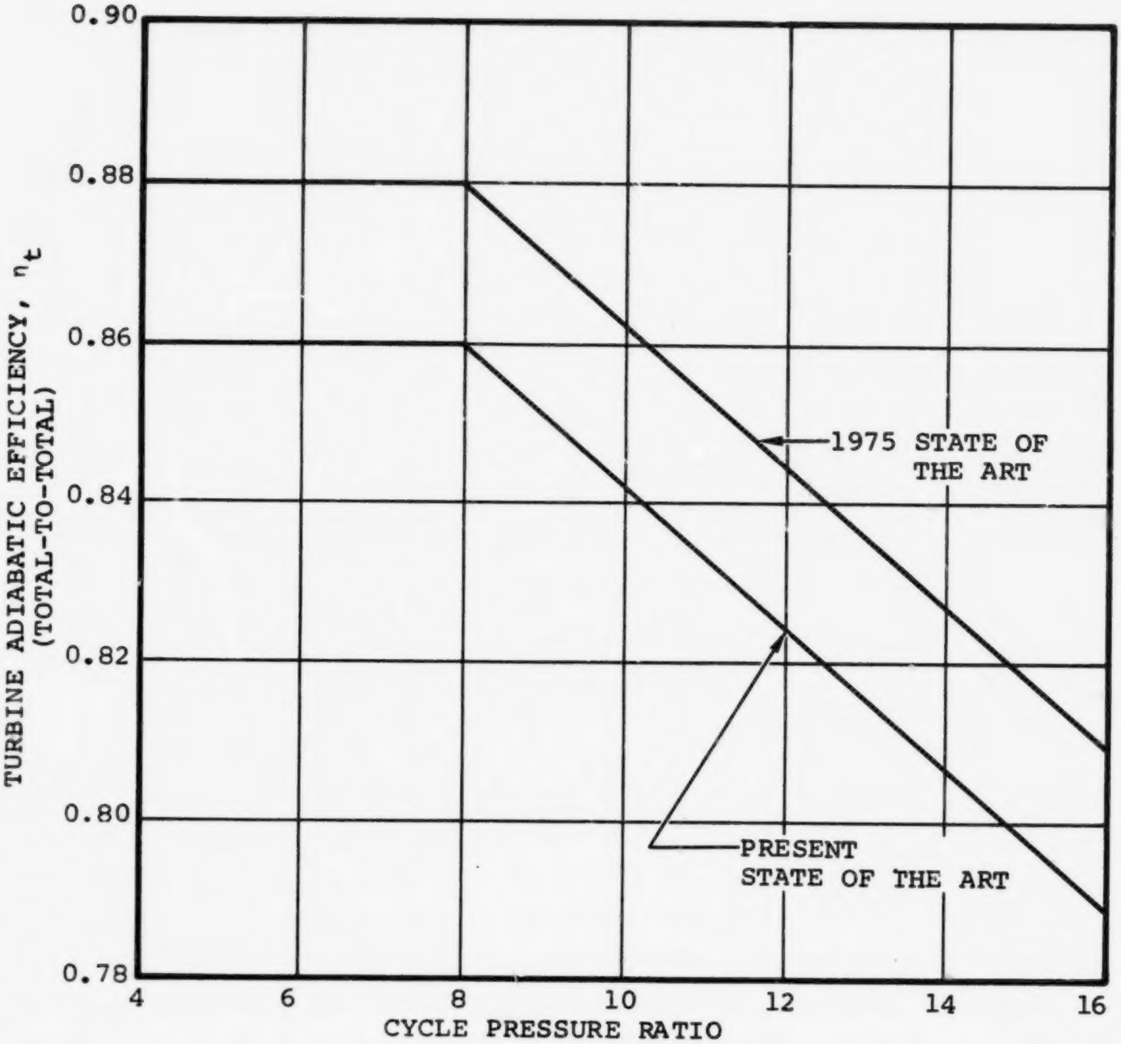
2.3.1 Current Technology

Selection of a turbine aerodynamic configuration is made on the basis of speed, work, and flow requirements dictated by engine cycle, compressor specific speed, and turbine stress considerations. The optimum turbine configuration requires suitable trade-off between aerodynamic performance and mechanical design; thus, there is no unique correlation of turbine efficiency as a function of cycle pressure ratio. Figures 2-8 and 2-9 show estimated efficiency levels that may be obtained for a single-stage axial, a two-stage axial, and a single-stage radial turbine as a function of cycle pressure ratio for engines in the 150-hp class.



SINGLE-AXIAL-STAGE TURBINE PERFORMANCE
(1 TO 2 LB/SEC ENGINE AIRFLOW)

FIGURE 2-8



TWO-STAGE AXIAL OR ONE-STAGE RADIAL TURBINE PERFORMANCE (1 TO 2 LB/SEC ENGINE AIRFLOW)

FIGURE 2-9

Quantitative radial and axial turbine efficiencies are estimated by theoretical analysis based on experimental data. These estimates, which include rotor blade and stator vane losses, are a function of the turbine detailed geometry. Therefore, these may be used to estimate performance penalties for nonoptimum aerodynamic designs, as well as estimates on performance improvements that might be realized from better materials or improved manufacturing techniques.

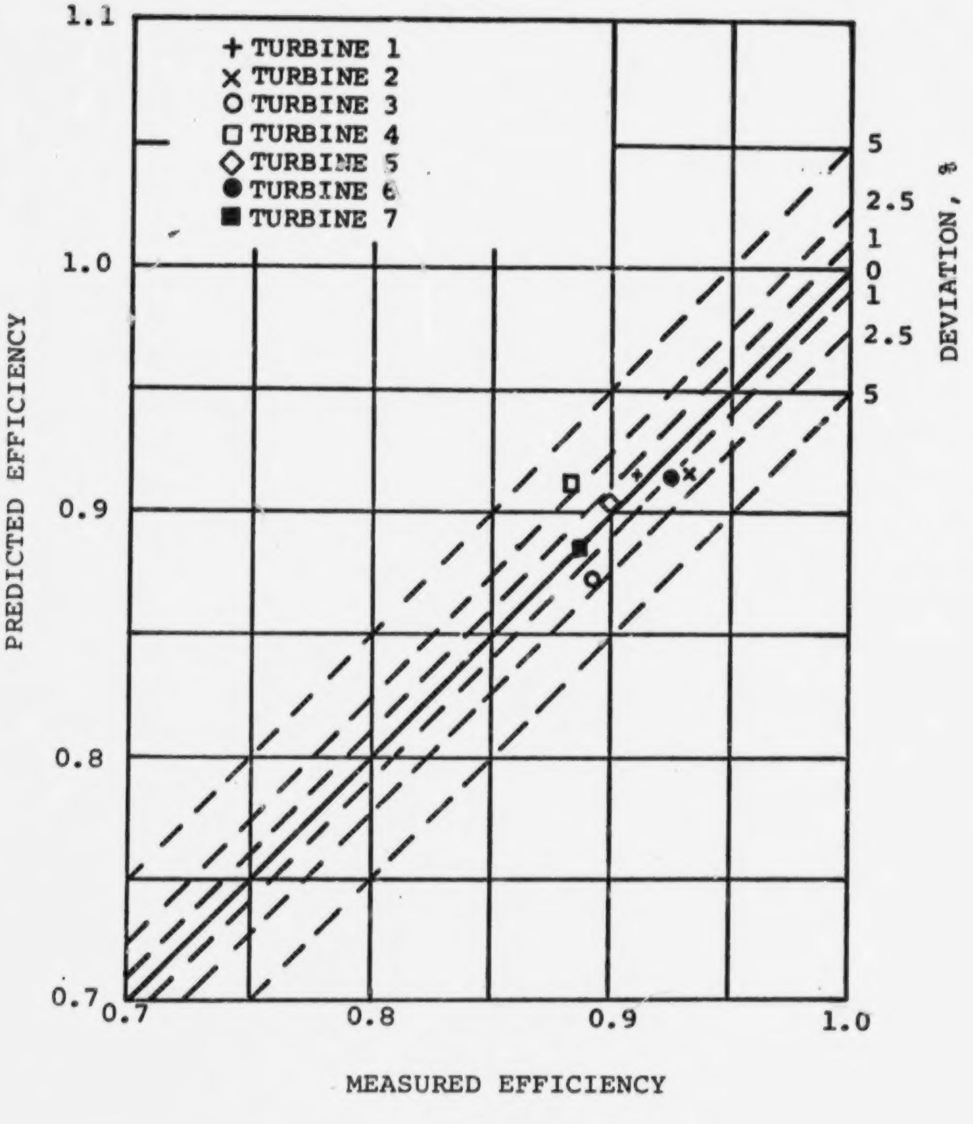
The efficiency-estimating technique yields the results shown in Figure 2-10 for seven AiResearch axial-flow turbines.

Figure 2-11 provides a comparison of the radial turbine efficiency prediction, as a function of specific speed, for various AiResearch radial turbines and for data available in the literature.

For small turbines of the size contemplated, radial units are usually more advantageous than axial (Figure 2-12). Efficiency comparisons (at zero-clearance) of radial and axial turbines for specific speeds in the range of interest demonstrate that actual radial turbine efficiency is higher, since clearance effects are less pronounced. In addition, radial turbines can operate at higher tip speeds than axial, thus permitting smaller diameters.

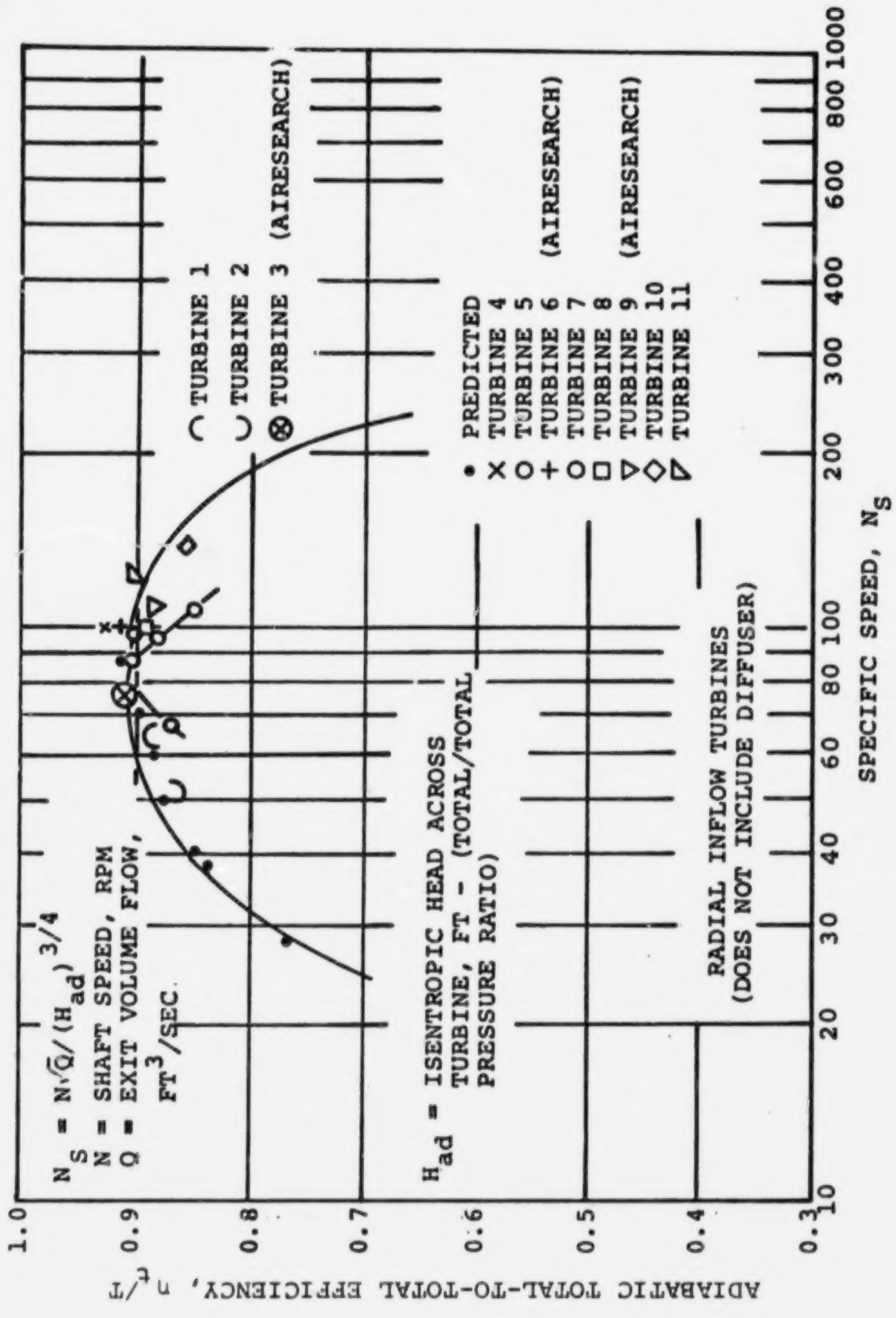
Radial wheels that are cast or forged can be run at substantially higher speeds than the axial-flow type, although the higher ultimate strength of forged wheels permits higher tip speeds than the cast wheels. Also, the higher the tip speed, the lower the relative blade temperature will be for a given gas total inlet temperature. Therefore, it is possible for stress-rupture life to increase with increased speed in spite of the higher stresses.

Other important advantages of radial flow turbines include higher stage work capability and resulting lower relative gas temperature. The low temperatures result in lower blade metal temperatures and,



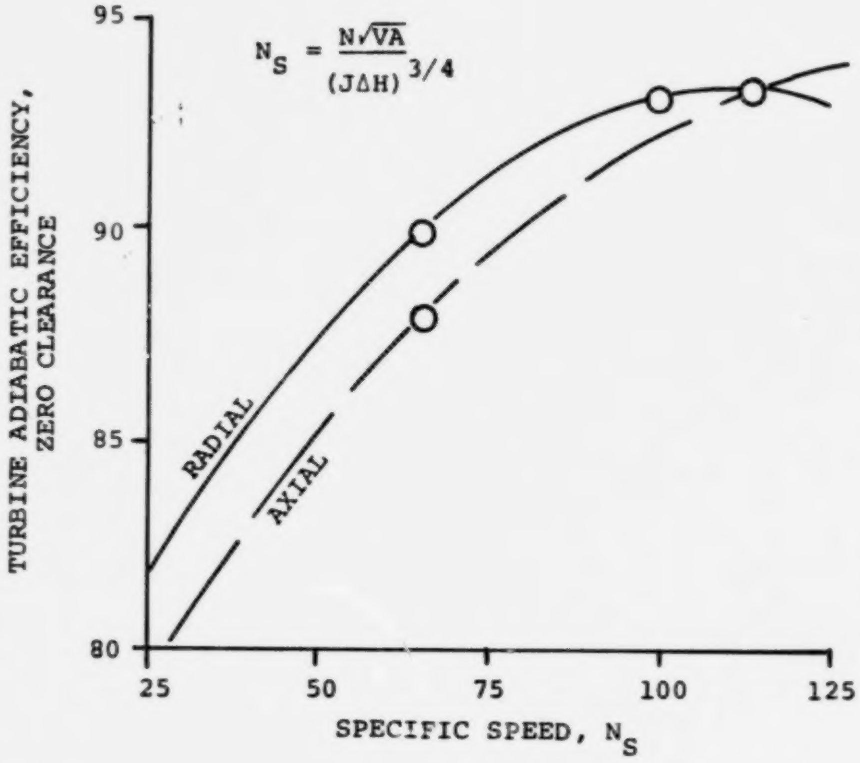
AIRESEARCH AXIAL TURBINE EFFICIENCY CORRELATION

FIGURE 2-10



EFFECT OF SPECIFIC SPEED ON RADIAL-INFLOW TURBINE EFFICIENCY

FIGURE 2-11



RADIAL TURBINE ADVANTAGES

1. HIGH TIP SPEED
2. GOOD EFFICIENCY, REDUCED CLEARANCE EFFECTS
3. HIGH-STAGE WORK, LOW RELATIVE GAS TEMPERATURES
4. RELATIVE TEMPERATURE DROPS WITH RADIUS
5. LOW RELATIVE MACH NUMBERS, TOLERANCE TO THICK BLADES AND COOLING MODIFICATIONS, LOW HEAT TRANSFER RATES

NOTES:

1. VA = EXIT VOLUME FLOW, FT^3/SEC
2. JΔH = ISENTROPIC HEAD ACROSS TURBINE, FT

AXIAL AND RADIAL TURBINE PERFORMANCE
(SMALL TURBINES)

FIGURE 2-12

hence, longer stress-rupture life. In addition, the blade relative temperature drops directly with the radius of a radial wheel. Consequently, at the exducer trailing-edge blade root where centrifugal stresses are at a maximum, the temperature is substantially lower than at the blade inlet. This is not true for an axial blade, as there is virtually no difference in relative blade temperature from inlet to exit. The radial turbine also has superior resistance to foreign object damage, usually has lower cost, and is generally more rugged.

The duty cycle imposed on the design has an all-important role in determining the relationship of temperature, speed, and size. Since the time at peak power in this application is minimal, stress-rupture life may not be as significant as in aircraft applications. In fact, mission analysis results show that the majority of operating time is spent at less than 50-percent power. In this case, low-cycle thermal fatigue from start cycles may be the critical failure mode for the gas generator turbine. The minimum design-life for analysis is specified as 10,000 starts.

2.3.2 1975 Estimated Technology

Turbine technology projections for 1975 include a 2-percentage-point improvement in gasifier turbine efficiencies and only 1-percentage-point increase in power turbine efficiencies, since the speed of this component may be selected to give the optimum aerodynamic configuration.

The improvement in efficiency levels is expected to be achieved by:

(a) Closer running clearance, utilizing more rigid structures and improved technology in shafting and bearing arrangements

(b) Contour control through improved manufacturing and quality control techniques at lower cost

(c) Improved analytical techniques

2.4 REGENERATORS/RECUPERATORS

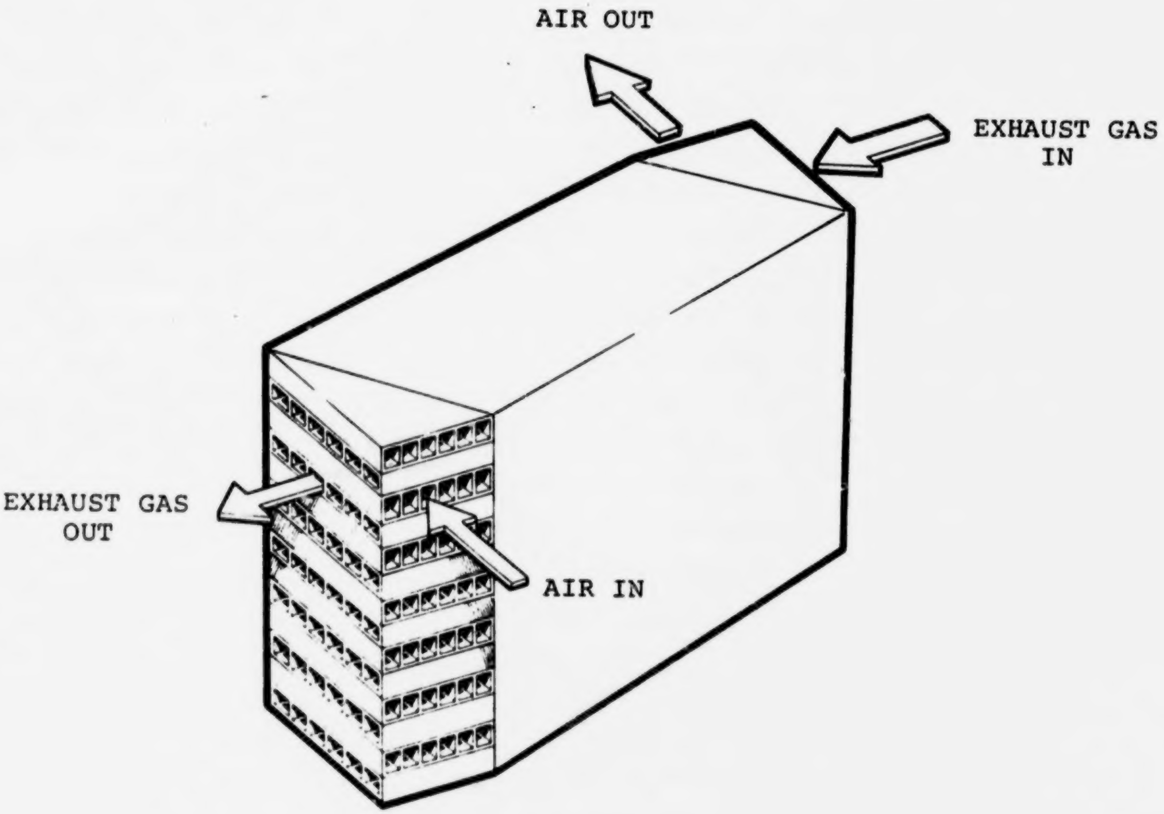
Some degree of waste heat recovery in the gas turbine engine is required to achieve a competitive specific fuel consumption at low output power for the automotive engine. Heat recovery methods considered in this study were:

(a) Fixed-boundary recuperator

(b) Rotary-regenerator

The fixed-boundary recuperator shown in Figure 2-13 is a static, direct-transfer heat exchanger, consisting of a matrix (core), headers, and plenums. The core may consist of either tubular, plate-fin metal sandwich or ceramic elements, depending on the design requirements. For high-effectiveness applications, counterflow configurations must be used.

The metal fixed-boundary recuperator has no moving parts, is backed with considerable field experience, and lends itself to modular construction--factors highly desirable for ease of installation and maintenance. Consideration of manufacturing cost and thermal stresses tends to favor a plate-fin counterflow design for the recuperator. However, it is difficult to package a unit designed for high effectiveness in a limited volume without incurring adverse duct losses and poor flow distribution. Nonuniform flow causes a degradation of effectiveness with excessive pressure drop.


AIR OUT
EXHAUST GAS IN
EXHAUST GAS OUT
AIR IN

TYPICAL PLATE-FIN PURE COUNTERFLOW RECUPERATOR

FIGURE 2-13

In establishing a recuperator configuration, the main factors to consider are the ability of the unit to withstand imposed steady-state and transient thermal gradients, the capability to transmit internal pressure loads, compatibility of the design to imposed vibration and shock loads, and ease of maintenance.

The plate-fin surfaces assumed for sizing the counterflow recuperator are production surfaces similar to those employed in heat exchangers for various commercial, military, and spacecraft applications. Offset rectangular strip-fins are used on both the air and gas sides. The offset feature of the fins yields high heat-transfer coefficients through recurring boundary-layer interruption and also contributes structural strength to the core. Plain rectangular fins are assumed in the triangular headers to minimize the pressure losses in the end sections of the core. The fin pitch in the headers is kept to a minimum, consistent with structural load requirements. The exhaust-gas and air-side fins are separated by thin plates that extend throughout the core.

In spite of more efficient manufacturing techniques, the cost of a metal plate-fin recuperator remains relatively high, largely because of the cost of the structural material itself. Also, there is a sharp upward break in the cost when switching from stainless steel to high-temperature alloys. Hence, a substantial increase in turbine inlet temperature must occur before the more expensive heat exchanger may be offset by a saving in fuel cost.

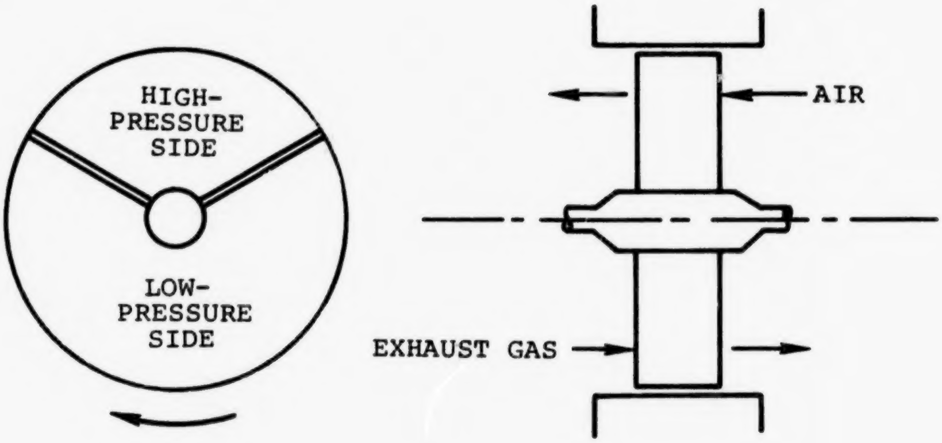
Low-expansion glass ceramic surfaces may prove to be an attractive alternative material for fixed-boundary recuperators. However, the present level of technology restricts the ceramic units to cross-flow configurations with comparatively larger core volume at a given thermal effectiveness.

Surface fouling, which is prone to occur with metallic cores at part-load conditions due to incomplete combustion, should be largely alleviated with better combustor and fuel control characteristics. Ceramic surfaces are more resistant to fouling due to the relative difference in thermal expansion coefficients between carbon deposits and ceramic materials.

In the rotary regenerator (Figure 2-14), heat is absorbed in the portion of the rotating matrix that is immersed in the hot exhaust gas and subsequently released to the compressor discharge air as a consequence of matrix rotation. Since the two gas streams flow through the matrix in opposite directions, a close approach to true counterflow conditions is possible. A mechanical drive is needed to achieve the continual, periodic movement of the heat-storage material between the exhaust gas and compressor discharge streams.

In the rotary generator, the matrix can be very compact, is relatively inexpensive, and is self-cleaning in operation. However, the complete regenerator requires power for matrix rotation and has seal-leakage that tends to be severe at high pressures. The penalty to cycle performance resulting from leakage may be largely offset, since it is practical to achieve a high design effectiveness in a rotary regenerator. The combined clearance and displacement leakage can be limited to an estimated 5 percent or less of compressor flow at a 4:1 pressure ratio, based on a 28-in.-diam Cercor disc.

The low expansion coefficient of the ceramic material imparts high thermal shock resistance and facilitates seal and bearing design. Low thermal conductivity helps to minimize the adverse effect of axial heat conduction through the matrix. Proper flow distribution should be more readily obtained in a rotary regenerator, since the core face areas are pie-shaped segments, in contrast to the high-aspect-ratio headers often required by a plate-fin recuperator.


HIGH-
PRESSURE
SIDE
LOW-
PRESSURE
SIDE
AIR
EXHAUST GAS

DISC-TYPE ROTARY REGENERATOR

FIGURE 2-14

Seal development represents a major problem in rotary regenerator technology, and little information on the subject is currently available from outside sources. The small allowable leakage-flow and large sealing perimeter necessitate the use of rubbing seals rather than controlled clearance seals. Seal leakage is smaller for one large disc than for two smaller ones with the same area. However, available disc size and packaging considerations may dictate a twin-disc configuration.

Glass-ceramic discs are uniquely well-suited for rotary regenerator service because of the following properties:

(a) Extremely low thermal expansion coefficient

(b) Low thermal conductivity

(c) Relatively low cost

(d) High heat capacity

2.5 TRANSMISSION AND DRIVE-LINE COMPONENTS

2.5.1 General

The following are desirable transmission characteristics:

(a) High efficiency over the normal operating range

(b) Low cost compatible with large-scale production

(c) Control simplicity for optimum performance

(d) Infinitely variable gear ratio for single-shaft engines

(e) Low volume and compact arrangement

(f) Low noise

(g) Low weight/power ratio

(h) Reverse-power and braking capability

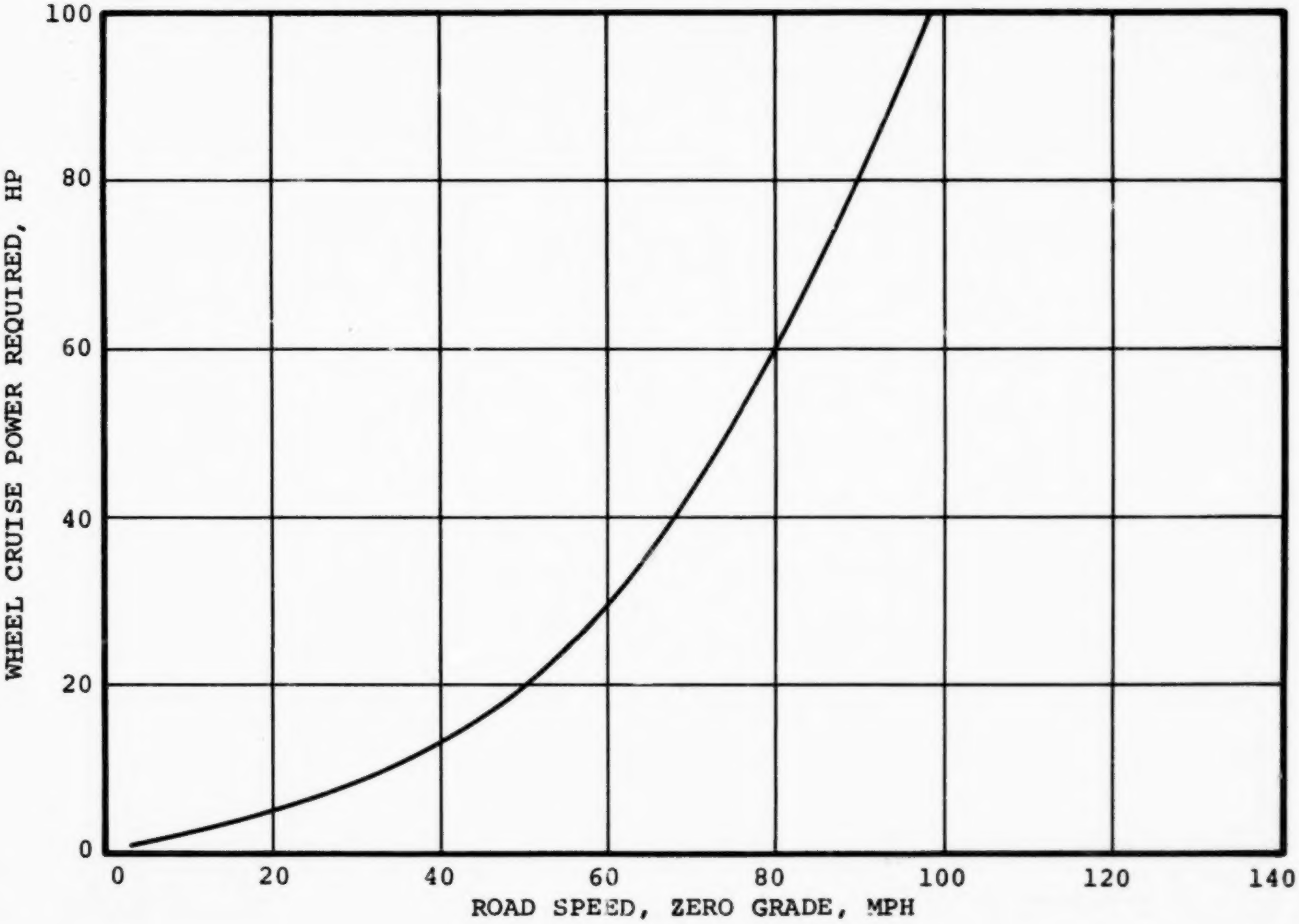
(i) Capability of absorbing road shocks

(j) Low power consumption during engine start and at idle

(k) Push-start capability

The transmission must be capable of supplying the relatively low power required in normal urban traffic at moderate or low cruise speeds with high efficiency. However, it must also be capable of operation at much higher powers and speeds without failure or large losses. Figure 2-15 illustrates the low cruise power required at low and moderate speeds for a 4000-lb automobile.

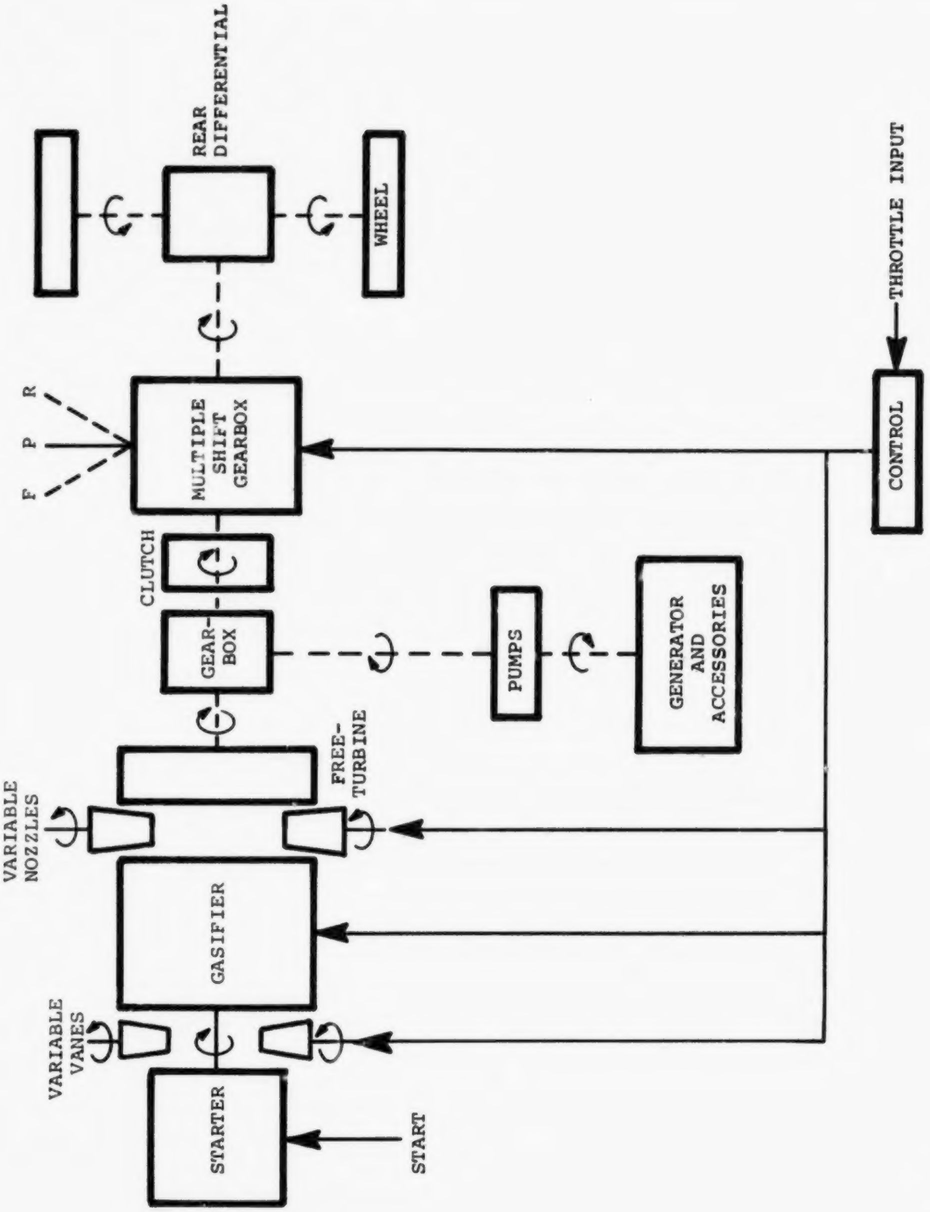
Transmission design will vary, depending on the type of gas turbine. A single-shaft unit requires an infinitely variable speed transmission to prevent engine stall in normal operation, whereas a free-turbine engine will use other types of transmissions, such as the conventional multiple-speed gearbox. Figures 2-16 and 2-17 illustrate typical engine-transmission configurations for the single-shaft and free-turbine engines, respectively. Certain transmission power losses exist in each. Thus, the losses involved in the primary gearbox, rear differential, and rear axle are substantially independent of the type of turbine. This is, of course, not true of the means of varying the speed ratio. The efficiency of a given transmission component depends upon both the power being transmitted and the operating speed. For example, the torque efficiency of a single gear at zero speed is essentially 100 percent, whereas it may easily have losses of 1 percent at speed. Studies of losses in gear sets indicate that losses may be divided into two types; those dependent on operating speed, and those dependent on input torque.



NOTE: 4000 LB AUTO
105°F DAY
OAP - "VEHICLE DESIGN GOALS" (APPENDIX 1)

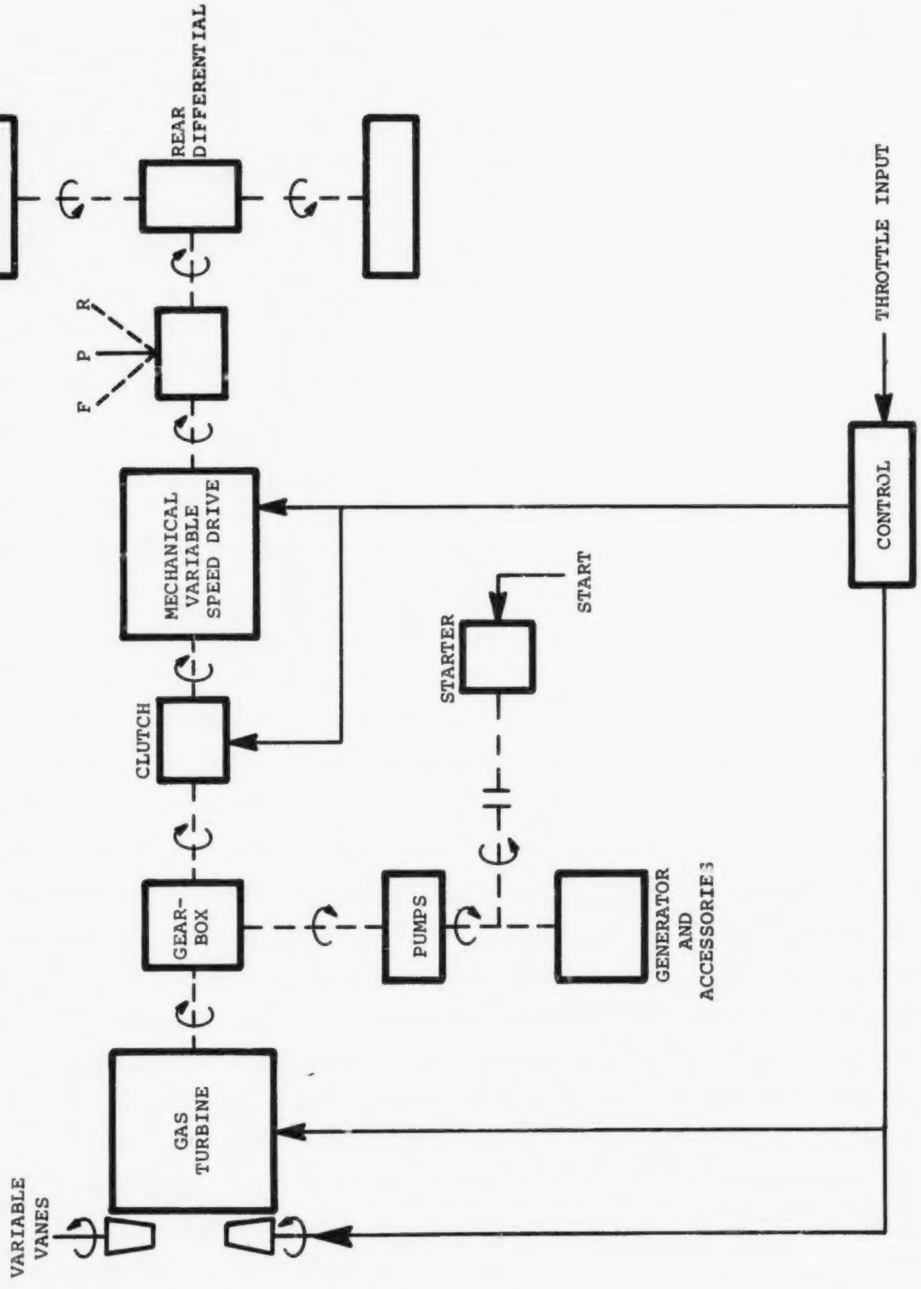
AUTOMOBILE CRUISE POWER REQUIREMENT

FIGURE 2-15


STARTER
START
VARIABLE VANES
GASIFIER
VARIABLE NOZZLES
FREE-TURBINE
GEAR-BOX
CLUTCH
MULTIPLE SHIFT GEARBOX
F
P
R
REAR DIFFERENTIAL
WHEEL
PUMPS
GENERATOR AND ACCESSORIES
CONTROL
THROTTLE INPUT

FREE-TURBINE DRIVE SCHEMATIC

FIGURE 2-16


WHEEL
REAR DIFFERENTIAL
F
P
R
MECHANICAL VARIABLE SPEED DRIVE
CLUTCH
GEAR-BOX
PUMPS
GENERATOR AND ACCESSORIES
STARTER
START
CONTROL
THROTTLE INPUT
GAS TURBINE
VARIABLE VANES

SINGLE-SHAFT GAS TURBINE DRIVE SCHEMATIC

FIGURE 2-17

2.5.2 Gearbox Losses

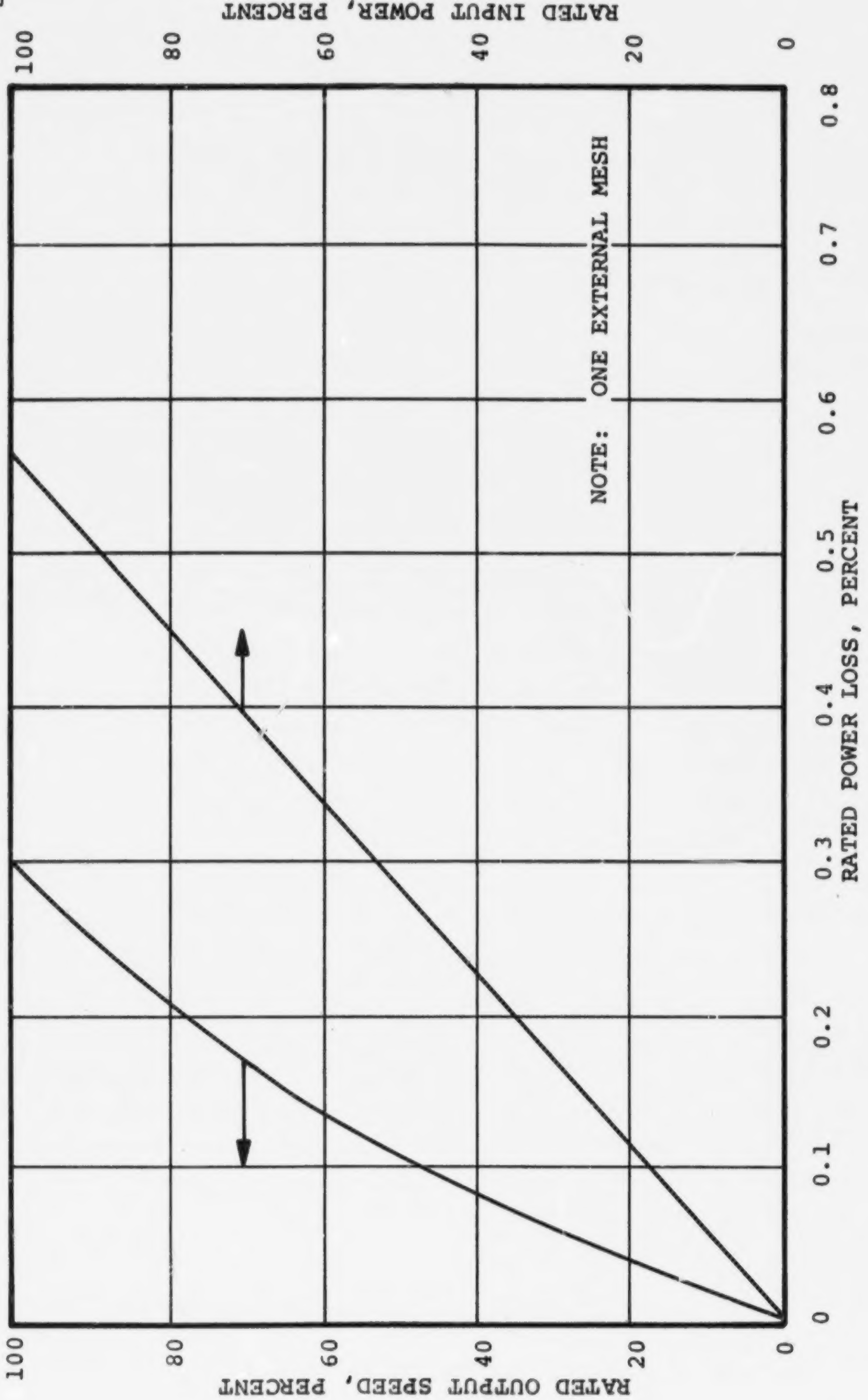
Losses in a conventional gearbox are of the three following types:

(a) Gear Mesh - Lubricated sliding loss proportional to torque, operating speed, tooth accuracy, and finish

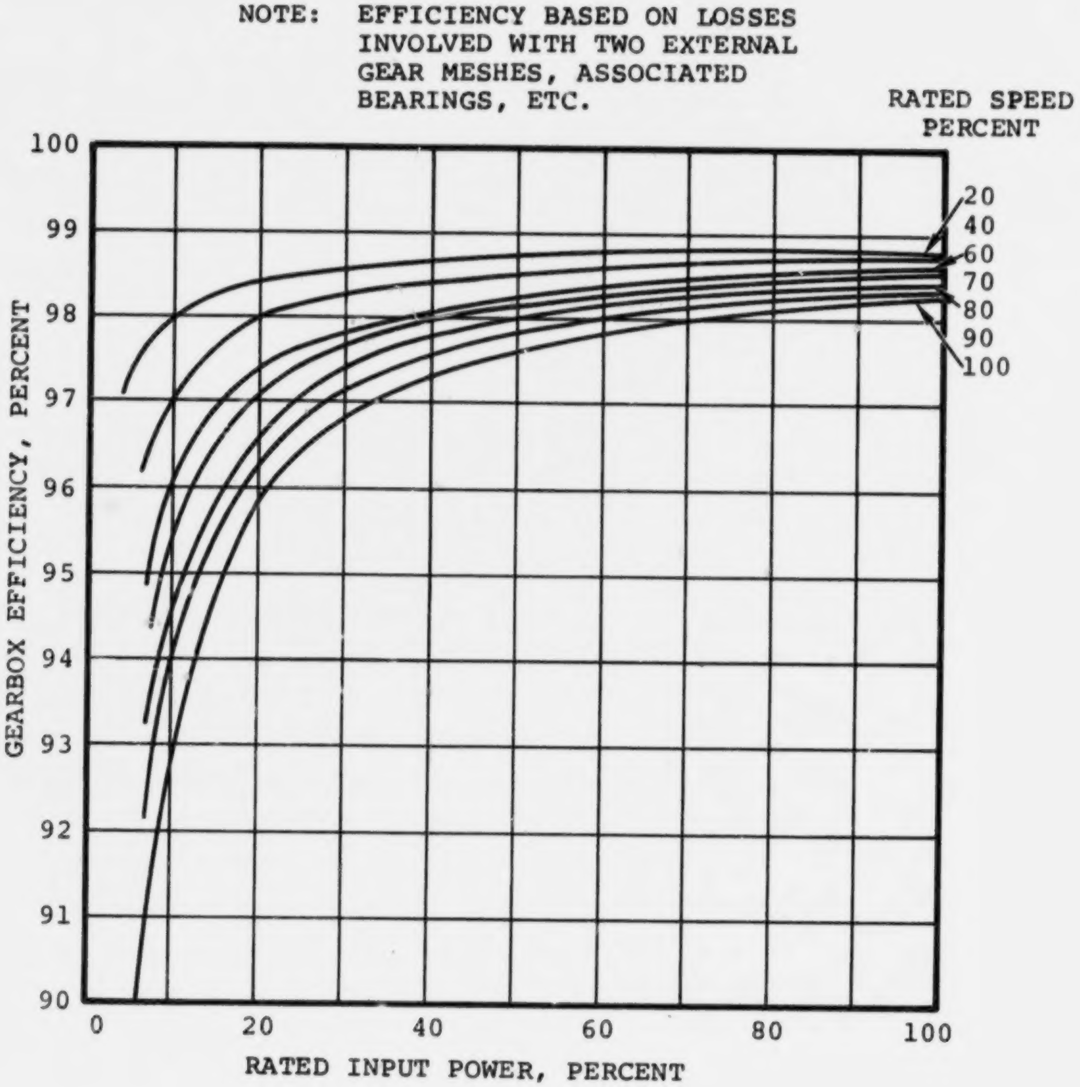
(b) Bearing - Proportional to torque, speed squared, and bearing size

(c) Lubricant Churning and Pumping - Proportional to the speed squared

The relationship of these losses in an aircraft gearbox involving two gear reductions, one external plus a planetary gear reduction, is given in Figure 2-18. Using a ratio of 0.80 for the two external gear meshes, relative to one external plus a planetary (one external plus one internal mesh), the percent power losses would remain at the first-stage reductions contemplated for the automotive gas turbine. This is consistent with the approximation often used in estimating gear losses of 1 percent per external mesh and 1/2 percent per internal mesh at rated power and speed. Using loss data in Figure 2-18, gear efficiency for the reductions involved in the proposed transmissions can be determined for the operating speed range. Transmission efficiency of the gas turbine primary gearbox is given in Figure 2-19. Losses in this gearbox are normally included in the calculation of the gas turbine output power to the transmission. Automobile rear differential efficiency has been derived and is illustrated in Figure 2-20; the combined efficiency of rear differential and axle is given in Figure 2-21 as a function of speed and power. The usual practice of using constant efficiency irrespective of speed and power for these components is obviously in error, particularly at higher speeds and lower powers. To illustrate, consider from Figure 2-15 that a cruise-level road speed of 60 mph requires 30 hp at the wheel. For a 150-hp



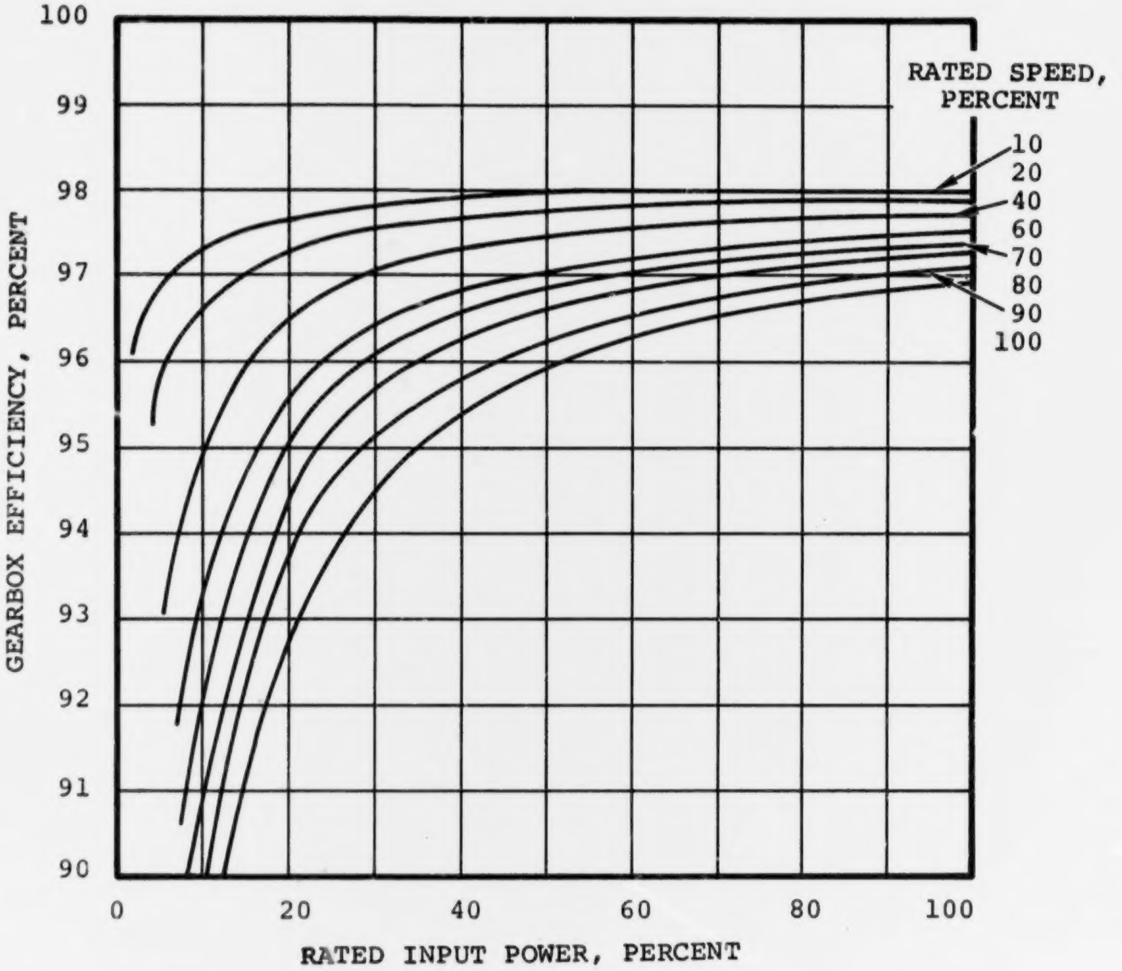
GEARBOX POWER LOSS
FIGURE 2-18



GAS TURBINE PRIMARY GEARBOX PERFORMANCE

FIGURE 2-19

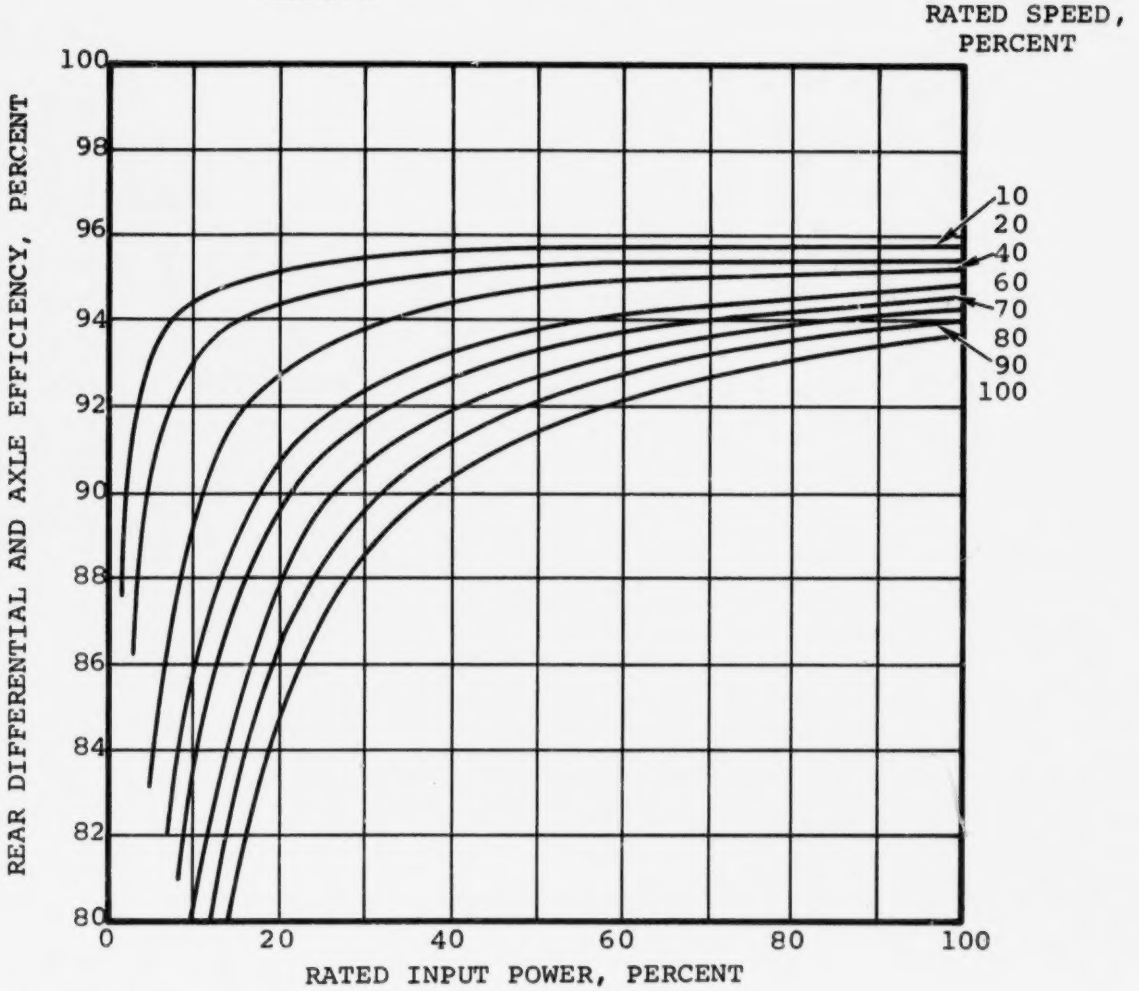
NOTE: EFFICIENCY BASED ON STANDARD OF 96.5 PERCENT RELATIVE TO A STANDARD OF 98.0 PERCENT IN PRIMARY GEARBOX



AUTOMOBILE REAR DIFFERENTIAL GEARBOX PERFORMANCE

FIGURE 2-20

NOTE: EFFICIENCY BASED ON STANDARDS OF 96.5 PERCENT FOR REAR DIFFERENTIAL AND 96.0 PERCENT FOR REAR AXLE, RELATIVE TO STANDARD OF 98.0 PERCENT IN PRIMARY GEARBOX



AUTOMOBILE REAR DIFFERENTIAL AND AXLE OVERALL EFFICIENCY

FIGURE 2-21

rated engine, the percent input power to the rear differential is about 22 percent, giving a rear differential and axle overall efficiency of 91 percent (taken from Figure 2-21). The constant efficiency approximation of 96.5 percent for the rear differential and 96.0 percent for the axle gives an overall efficiency of 92.6 percent. Maximum power acceleration gives rear differential and axle overall efficiencies of approximately 96 to 93.5 percent, depending upon speed, which is larger than the constant 92.6 percent.

2.5.3 Transmissions

There are four basic transmission types, defined as the mechanism of the drive line that changes the engine-to-wheel speed ratio (depending on speed and power) that have been studied for the gas-turbine-powered automobile:

(a) Multiple-shift mechanical gear

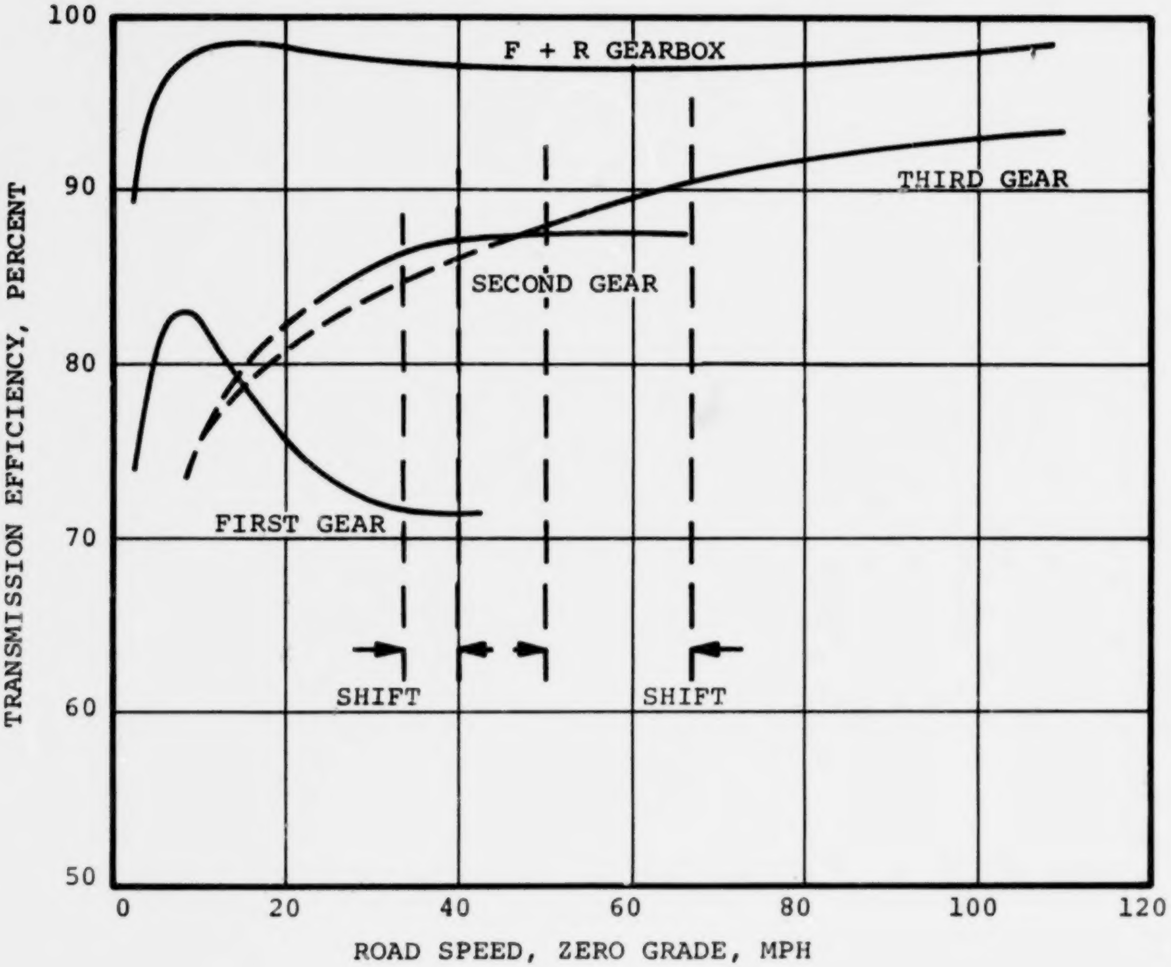
(b) Electric drive (generator, motor, and controls)

(c) Hydromechanical or hydrostatic power-splitting

(d) Infinitely variable speed ratio (toroidal drive and belt drive)

The very low power required by the automobile at normal urban cruise speeds (Figure 2-15) emphasizes the need for a highly efficient transmission. Efficiencies of a typical current automobile three-speed automatic gearbox and a typical forward and reverse (F&R) gearbox are shown in Figure 2-22. These efficiencies are plotted versus vehicle speed in mph, assuming the cruise powers in Figure 2-15. Figure 2-22 illustrates the simple three-speed gearbox transmission efficiencies between 80 and 90 percent in the normal urban driving speed range.

NOTE: "B" AUTO TRANSMISSION POWER, AS REQUIRED BY A 4000-LB AUTO (DATA SUPPLIED BY OAP)



AUTOMOBILE THREE-SPEED GEARBOX PERFORMANCE

FIGURE 2-22

While a multiple-speed gearbox can be incorporated with a free-turbine engine, it is necessary to consider more complicated transmissions for the single-shaft gas turbine because of poor torque characteristics. In the past, gas turbine-generator combinations have been considered in which the electrical output was supplied to a motor driving the wheels. While numerous unique arrangements were developed, the overall efficiencies of such drives remained unreasonably low and the overall potential cost unreasonably high. Table 2-1 gives typical data on electric motors from which the best possible transmission efficiency (in the order of 77 percent) is derived:

$$\eta_t = \eta_g \eta_m \eta_c = (0.99)\ (0.90)\ (0.95) = 0.77$$

where:

η_t = transmission efficiency

η_g = generator efficiency

η_m = motor efficiency

η_c = control efficiency

Also, the potential cost of such a system was quite high due to motor size and control complexity. One survey[1*] indicated that controls would cost from $2.00 to $5.00/hp--probably much too high to meet the 110-percent cost-of-ownership requirement but indicative of the problems involved. Much lower electrical transmission efficiencies could be expected at the low part-power operation. Consequently, no further study was made of the electrical transmission system.

Studies of hydromechanical transmissions have been extensive in the last 10 years and have provided improvement in the concepts

---

*Superscript numbers refer to References in Section 13.

TABLE 2-1

DC-SERIES VEHICLE MOTORS[2]

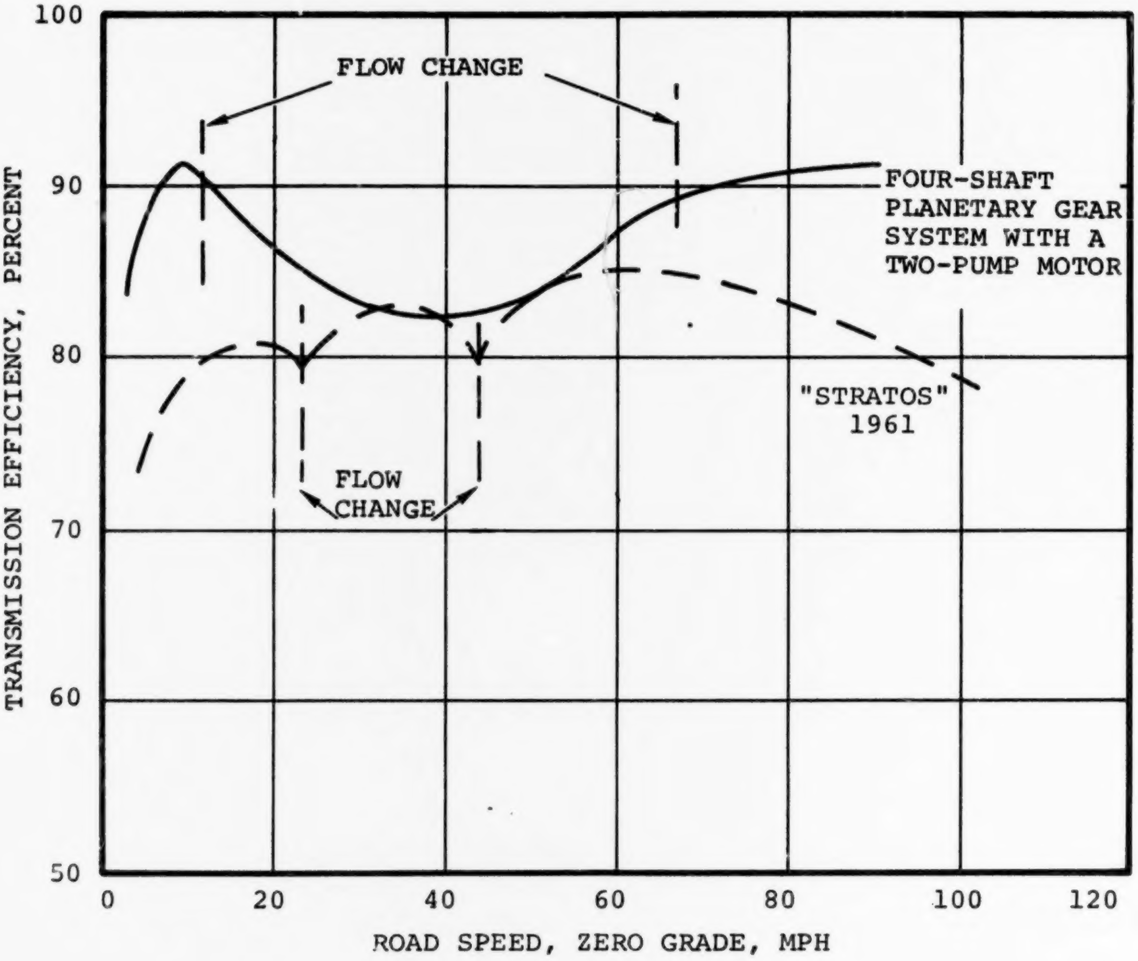
Property	Unit	No. 1	No. 2
Continuous Power	hp	11	40
Intermittent Power	hp	25	100
Speed	rpm	3200	8000
Efficiency	%	84	90
Voltage	vdc	72	120
Cooling		Open, fan	Blower
Length	in.	18-1/2	15-1/2
Diameter	in.	9-1/2	9-3/8
Weight	lb	165	160

advanced and in reductions in the size and weight of components. An early hydromechanical transmission, proposed in 1961 at Stratos, had an efficiency curve as shown in Figure 2-23, which also includes a more recent and more optimum four-shaft system[3]. However, hydromechanical systems suffered from a low-transmission efficiency in the normal operating regime and required a substantial amount of hydraulic system components. Figure 2-24 shows the typical present-day efficiency changes with a decrease in power. Although hydromechanical transmission could be adapted for the single-shaft gas turbine, new variable-speed mechanical drives offered greater potential because of higher efficiency in the normal operating range and lower production costs. As a result, major consideration was given to two such infinitely variable mechanical drives--the toroidal drive of TRACOR, Inc., and a novel belt drive invented at AiResearch.

The toroidal drive has had a long history of development[4] and is currently highlighted by a resurgence of interest, based on better traction fluids[5] and greater understanding of the operation. The belt drive benefits from recent developments in high-strength rubberized composites (such as the Gates Rubber Co., polyflex belt) and is based on a unique bent-axis concept that affords nearly optimum design. Estimated constant-speed, road-load efficiencies for these drives are given in Figure 2-25, and more detailed efficiency curves with respect to various powers and speed ratios are shown in Figures 2-26 and 2-27. Both belt and toroidal drives have an operating speed ratio range of 6.0, although the belt drive operates at lower absolute speeds than the toroidal. This belt drive has computed efficiencies significantly higher than any other transmission, but the new proprietary concept has not yet been proven experimentally.

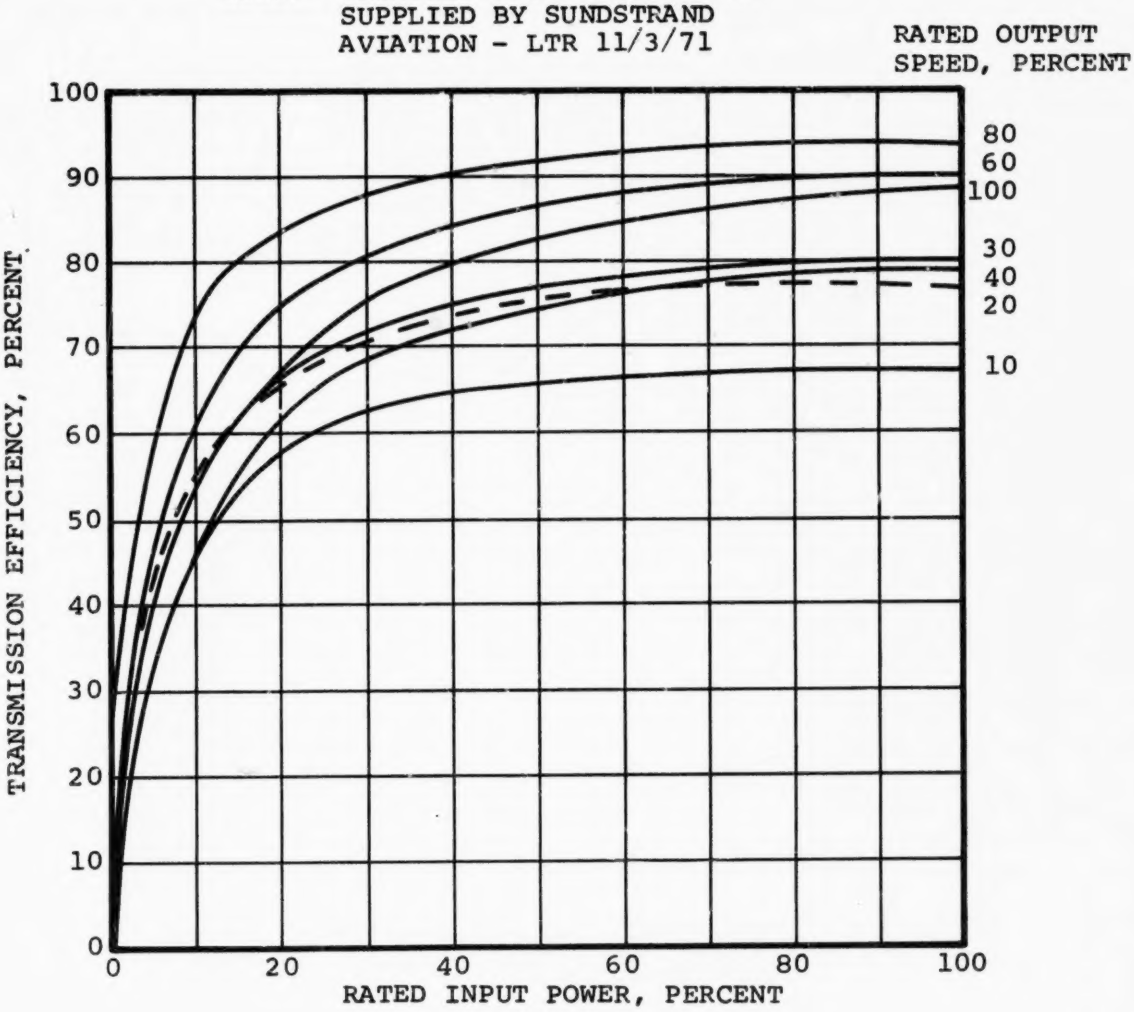
When comparing figures, the data for maximum power transfer over the speed range is shown only in Figure 2-23, whereas Figures 2-22 and 2-25 present cruise power at the speeds involved for a 4000-lb automobile. The efficiencies of Figure 2-23 would decline significantly

NOTE: MAXIMUM OUTPUT POWER ONLY



PERFORMANCE OF HYDROMECHANICAL TRANSMISSION
FOR GAS TURBINES

FIGURE 2-23

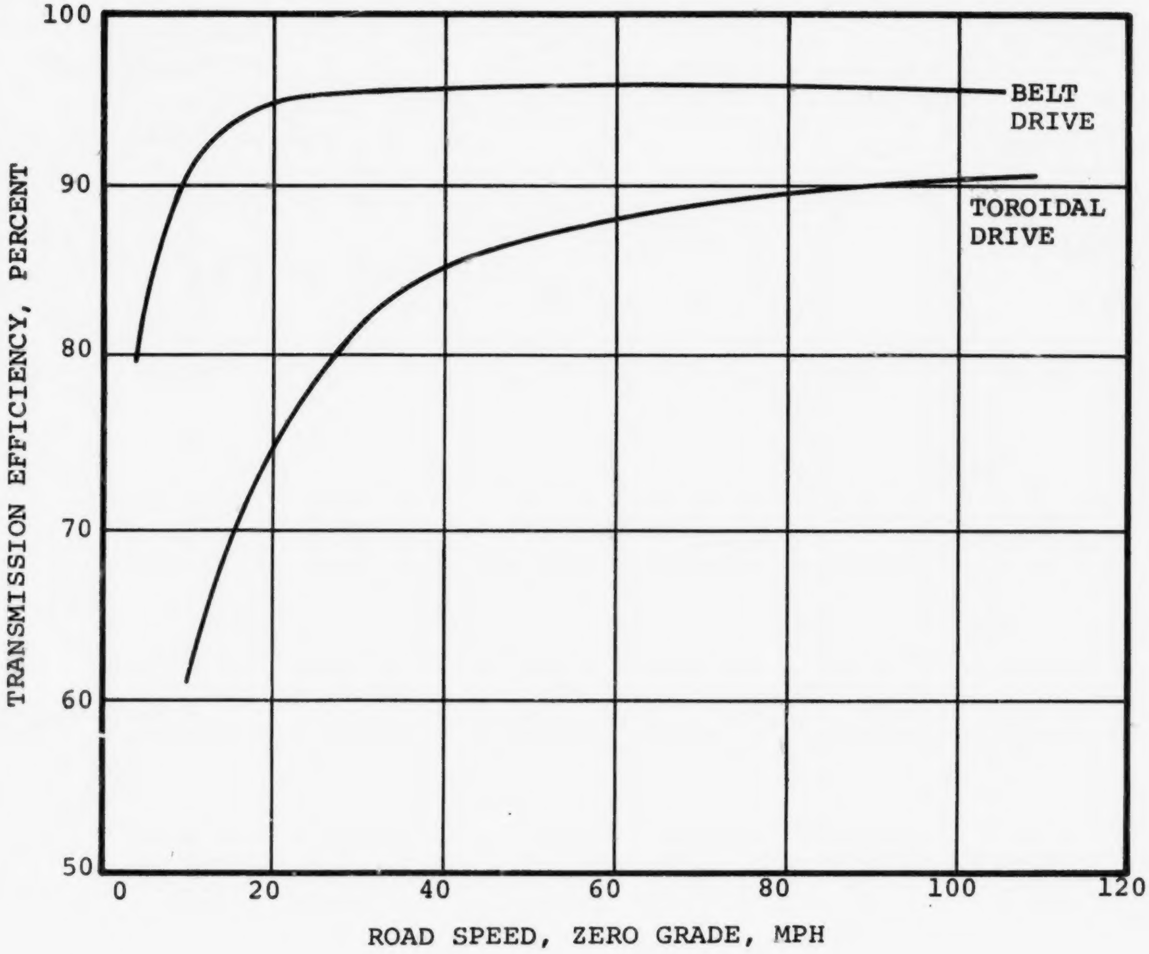


NOTE: OBTAINED FROM INFORMATION SUPPLIED BY SUNDSTRAND AVIATION - LTR 11/3/71

HYDROMECHANICAL INFINITELY VARIABLE RATIO TRANSMISSION PERFORMANCE

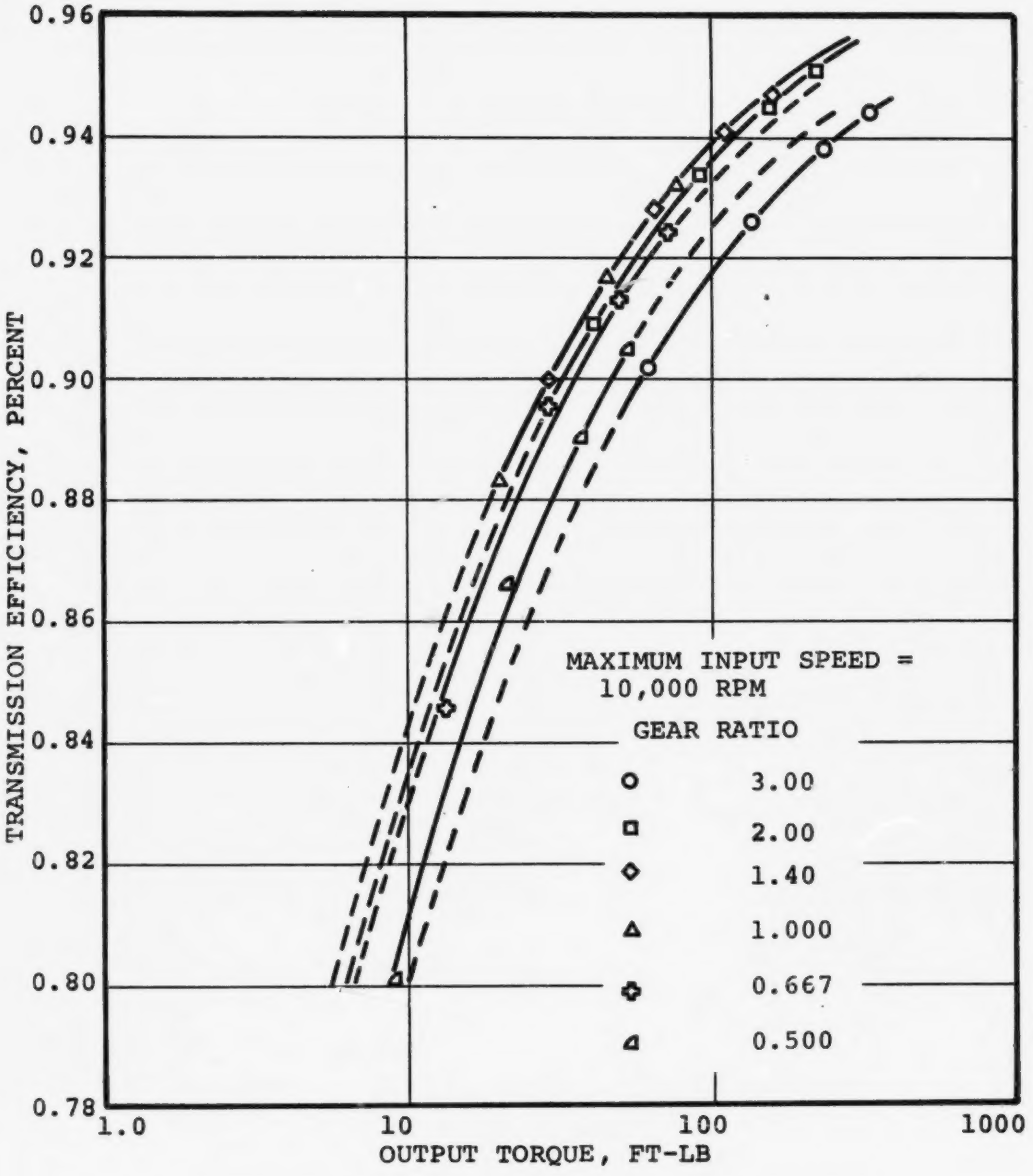
FIGURE 2-24

NOTE: POWER AS REQUIRED BY 4000-LB AUTO INCLUDES ONE EXTERNAL GEAR MESH LOSS ON EACH DRIVE



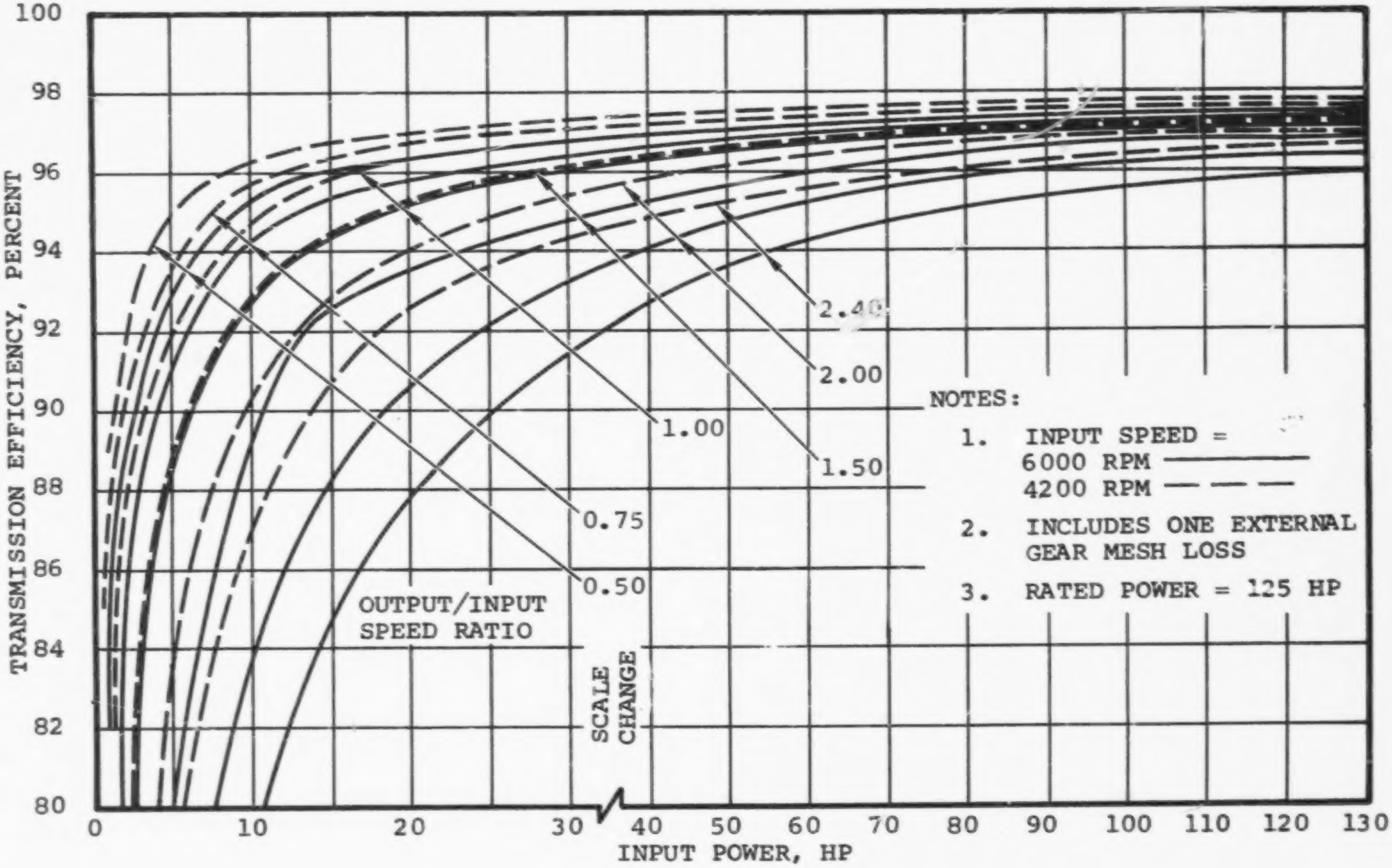
PERFORMANCE OF INFINITELY VARIABLE SPEED MECHANICAL TRANSMISSIONS

FIGURE 2-25



TOROIDAL VARIABLE SPEED DRIVE PERFORMANCE

FIGURE 2-26



VARIABLE SPEED BELT DRIVE PERFORMANCE

FIGURE 2-27

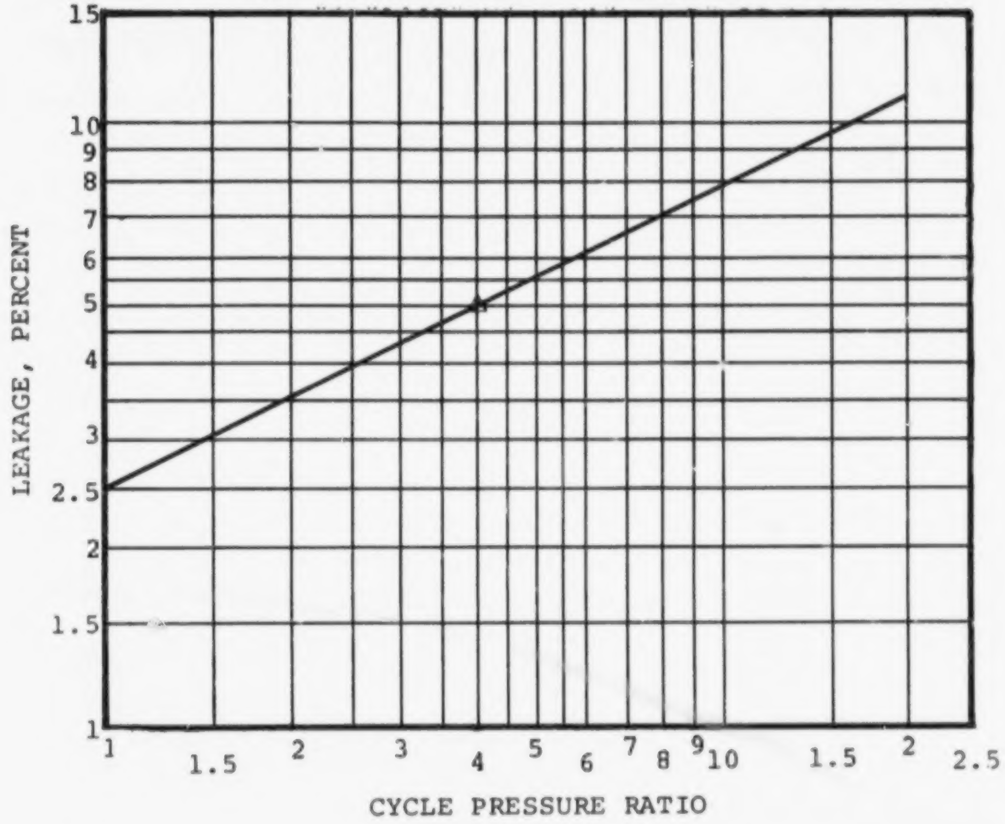
by using the lower cruise powers required at the lower speeds shown in Figure 2-24. Toroidal drive efficiencies largely overlap those available from the three-speed gearbox.

2.6 ENGINE PARASITIC LOSSES

To accurately predict the performance of an engine, it is necessary to include two types of elements in the computer mathematical model of the engine--engine component performance maps and parasitic losses. Both are determined by actual test data. The parasitic losses include leakage and pressure losses, cooling flow requirements, and mechanical inefficiencies for the system components.

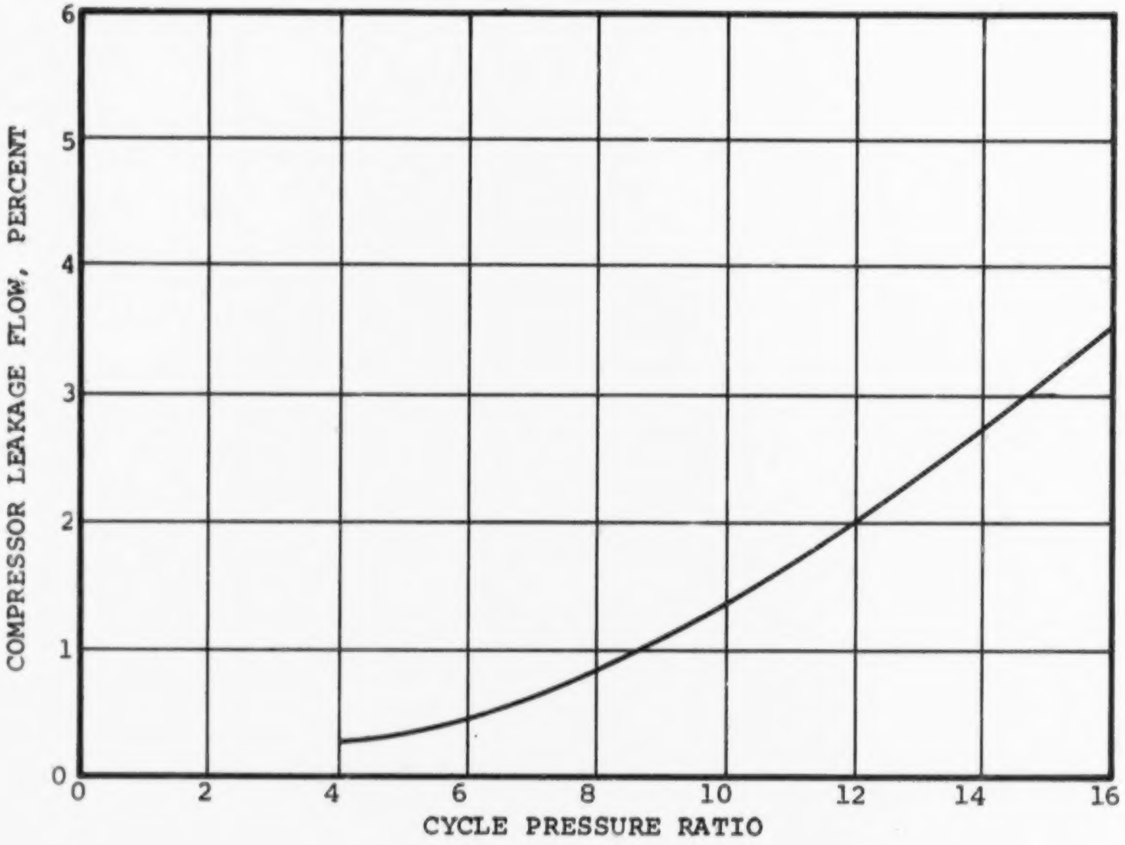
The regenerator seal leakage flow is a function of cyclical pressure ratio (Figure 2-28); the cycle leakage flow is also a function of pressure (Figure 2-29). The cooling required for the turbine is a function of the gasifier turbine inlet temperature, as evident in the curve of Figure 2-30. Table 2-2 lists the typical losses at design-point (100-percent power condition) for an engine in this power class.

NOTE: Δ ESTIMATED FROM PREVIOUS STUDY INVOLVING CERCOR J-1728-0-72, 28-IN.-DIAM DISC WITH AN EFFECTIVE SEAL CLEARANCE OF 0.00188 IN.



REGENERATOR SEAL LEAKAGE DESIGN GOALS

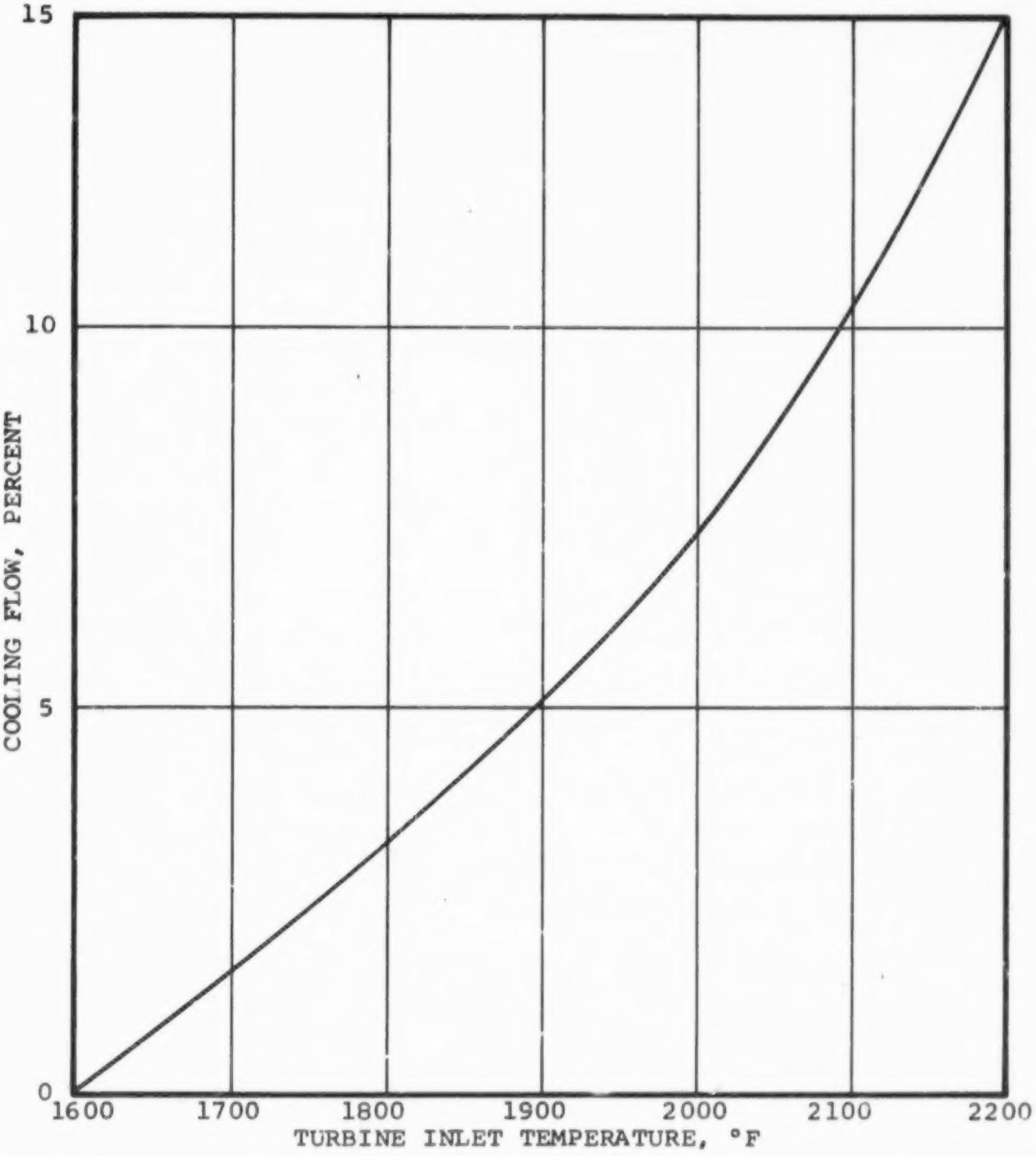
FIGURE 2-28



COMPRESSOR LEAKAGE LOSS

FIGURE 2-29

70



TURBINE COOLING FLOW REQUIREMENTS

FIGURE 2-30

TABLE 2-2

ENGINE PARASITIC LOSSES

Parameters	Unit	Loss
Accessories and Spin Losses		
Gasifier shaft, free-turbine cycle	hp	5.0
Power turbine shaft, free-turbine cycle	↓	1.5
Single-shaft		6.0
Power extraction efficiency	%	2.0
Cycle Pressure Losses	↓	
Inlet		1.5
Exhaust		1.0
Compressor discharge		1.0
Turbine transition duct		2.0
Combustor Pressure Loss		4.0
Heat Exchangers Pressure Losses		
Recuperator (core plus duct)		
Air side		2.3
Gas side		6.7
Regenerator		
Air side		2.3
Gas side		6.7
Power Turbine Pressure Loss		
Diffuser total		5.0

SECTION 3

PARAMETRIC DESIGN-POINT CYCLE STUDIES

A parametric design-point cycle study was made of several single-shaft and free-turbine engines. Design-point variables considered in the study included the following:

(a) Cycle pressure ratio - 4.0 to 16.0

(b) Turbine inlet temperature - 1600° to 2200°F

(c) Component efficiencies - function of turbine inlet temperature, cooling techniques, and component size

The parametric study was made with the component efficiencies established for 1971 and 1975 technology levels for a gas turbine engine having a through-flow rate of 1 to 2 lb/sec. Because the required cooling flow varies with turbine size, turbine inlet temperature, and cooling techniques, the parametric study was conducted using three cooling-flow rates for each cycle configuration; 0, 50, and 100 percent, as defined in Figure 2-30, Section 2. The 100-percent cooling is representative of the amount used for aircraft gas turbine applications. By supplying these data, with other basic information specified in Figure 3-1 for a particular engine cycle, a computation of engine performance was obtained by computer analysis, using a verified, existing program. Thus, "carpet plots" of specific fuel consumption and specific powers were obtained for a given amount of cooling flow as a function of lines of constant cycle pressure ratio and turbine inlet temperature on a sea-level, standard day.

This information was used for a given cycle to make qualitative judgements regarding the selection of cycle pressure ratio and temperature. Based on this choice, the compressor and turbine configurations

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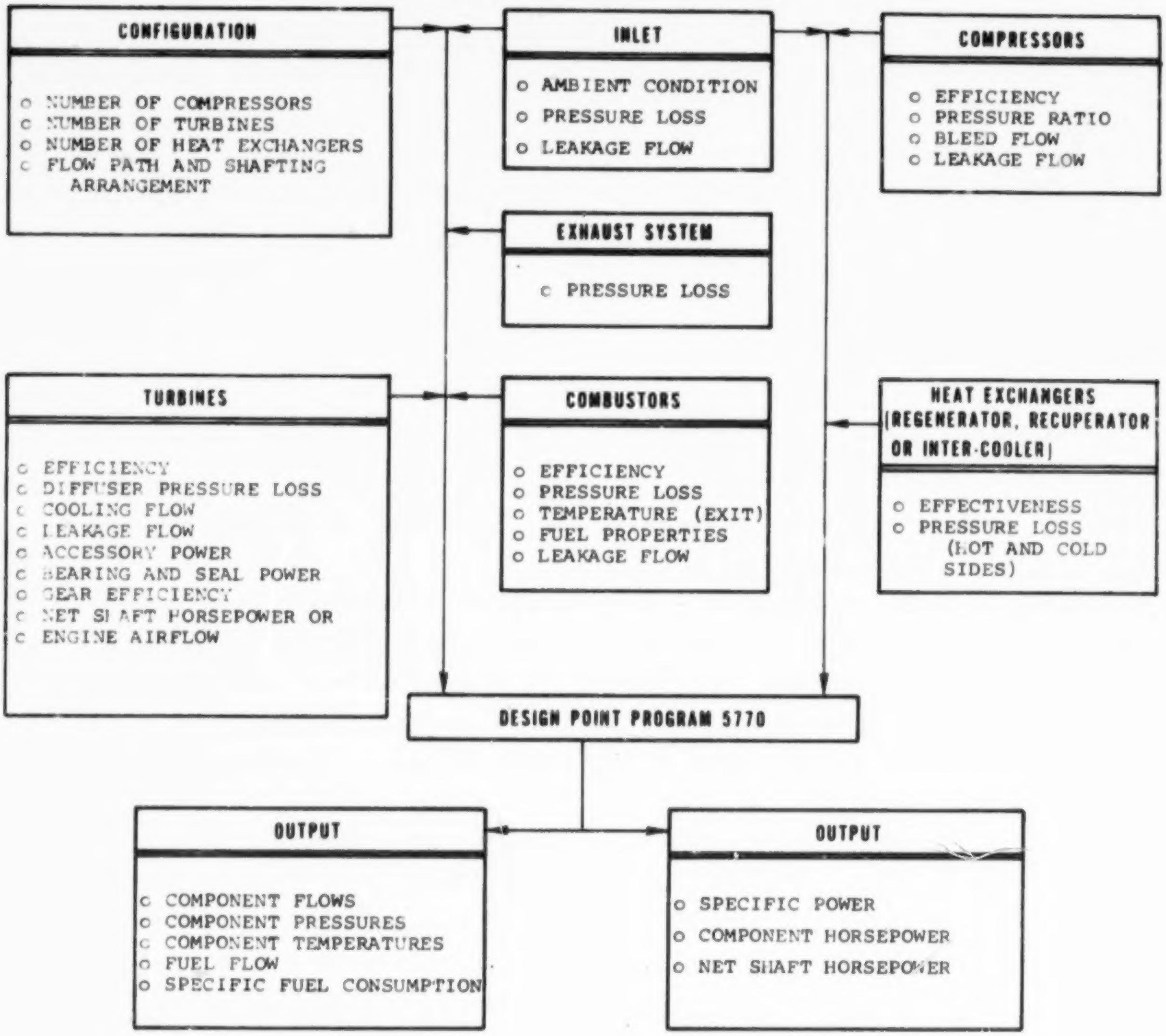

CONFIGURATION
o NUMBER OF COMPRESSORS
o NUMBER OF TURBINES
o NUMBER OF HEAT EXCHANGERS
o FLOW PATH AND SHAFTING ARRANGEMENT
INLET
o AMBIENT CONDITION
o PRESSURE LOSS
o LEAKAGE FLOW
COMPRESSORS
o EFFICIENCY
o PRESSURE RATIO
o BLEED FLOW
o LEAKAGE FLOW
EXHAUST SYSTEM
o PRESSURE LOSS
TURBINES
o EFFICIENCY
o DIFFUSER PRESSURE LOSS
o COOLING FLOW
o LEAKAGE FLOW
o ACCESSORY POWER
o BEARING AND SEAL POWER
o GEAR EFFICIENCY
o NET SHAFT HORSEPOWER OR
o ENGINE AIRFLOW
COMBUSTORS
o EFFICIENCY
o PRESSURE LOSS
o TEMPERATURE (EXIT)
o FUEL PROPERTIES
o LEAKAGE FLOW
HEAT EXCHANGERS (REGENERATOR, RECUPERATOR OR INTER-COOLER)
o EFFECTIVENESS
o PRESSURE LOSS (HOT AND COLD SIDES)
DESIGN POINT PROGRAM 5770
OUTPUT
o COMPONENT FLOWS
o COMPONENT PRESSURES
o COMPONENT TEMPERATURES
o FUEL FLOW
o SPECIFIC FUEL CONSUMPTION
OUTPUT
o SPECIFIC POWER
o COMPONENT HORSEPOWER
o NET SHAFT HORSEPOWER

DESIGN POINT CYCLE PROGRAM

FIGURE 3-1

were selected that best provided the desired rated (and anticipated part-load) specific power and specific fuel consumption. In addition, cycle complexity, ruggedness, reliability, and component costs were qualitatively considered in the selections.

Once the preliminary candidate cycles were selected, the following additional activities were accomplished:

(a) Compressors and turbines were sized.

(b) Cooling flow requirements were established to give the required engine life for the selected materials, speeds, and maximum cycle temperature.

(c) Compressor and turbine rotor speed compatibility was determined with respect to efficiency level compromise, as imposed by turbine stress limitations.

For the purpose of improving low part-load specific fuel consumption, the following cycle modifiers were investigated during the parametric design study:

(a) Variable area power turbine nozzles

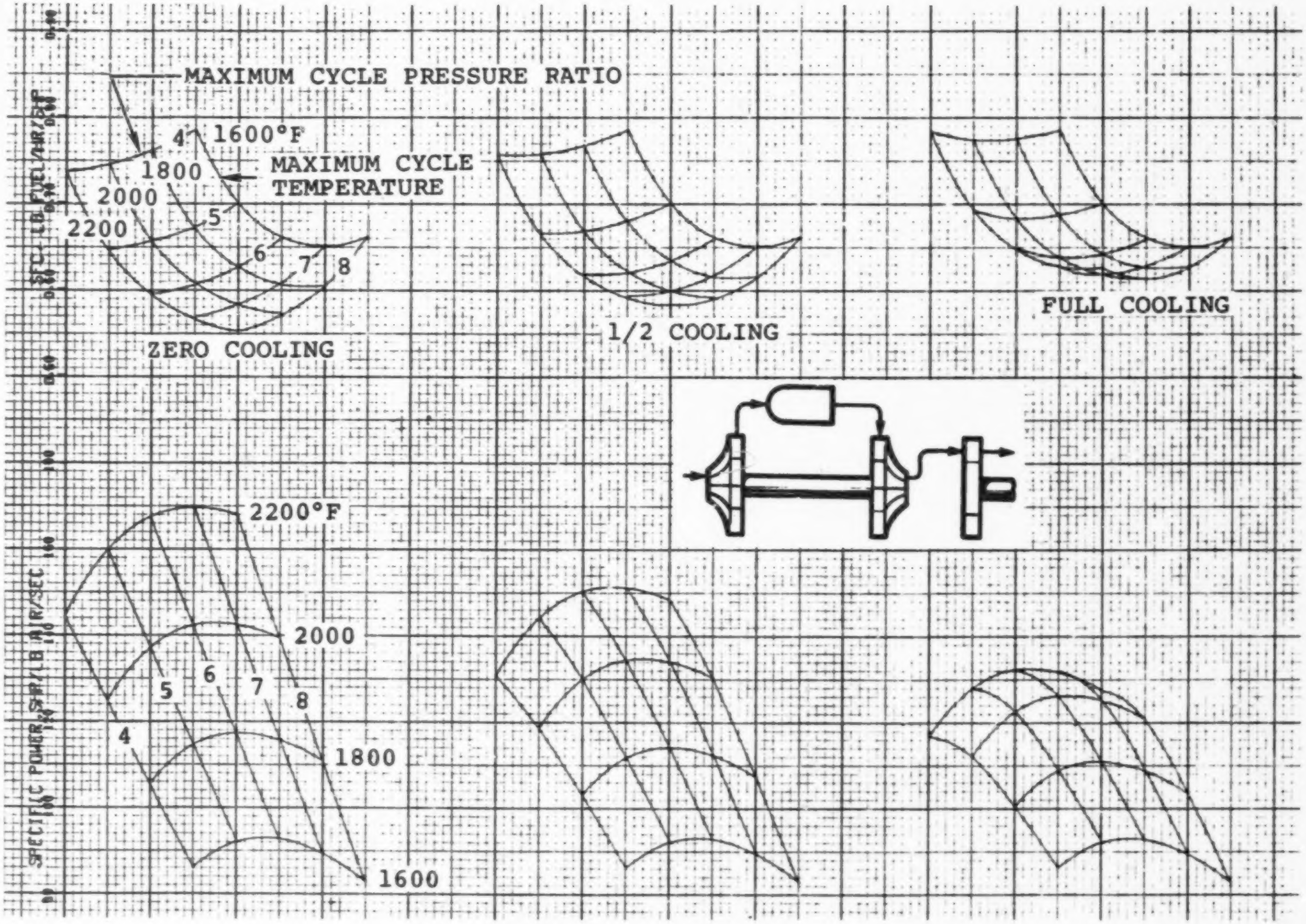
(b) Heat exchangers of two types:

    (1) Recuperator (fixed boundary)

    (2) Regenerator (rotary)

Several additional cycle modifiers were considered during off-design performance analysis (Section 4).

Figures 3-2 through 3-9 show carpet-plots of design-point performance for the selected preliminary candidate gas turbine cycles.



NON-REGENERATED FREE-TURBINE CYCLE
SEA-LEVEL, 59°F DAY, 150 SHP

FIGURE 3-2

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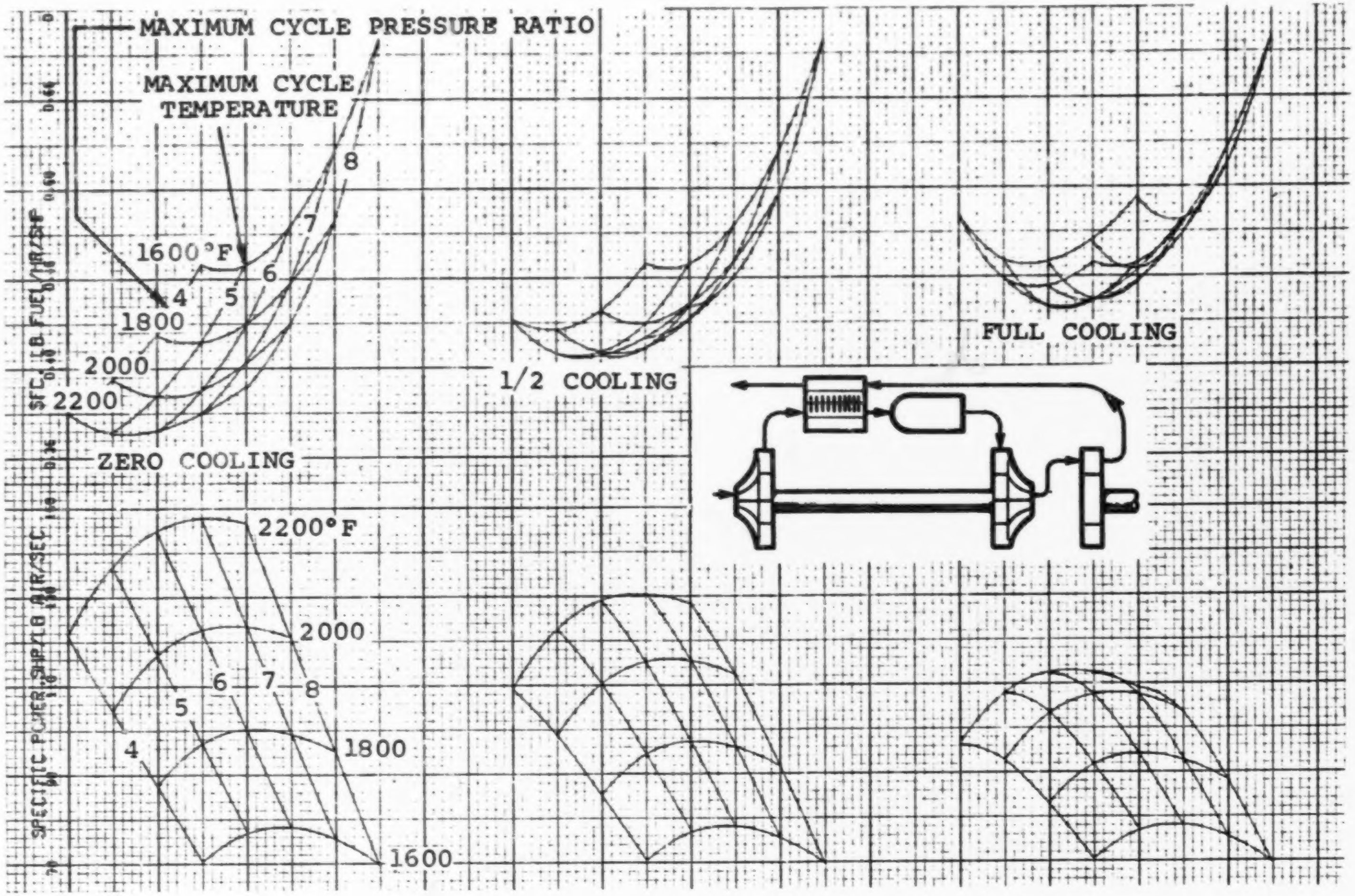

RECUPERATED FREE-TURBINE WITH FIXED GEOMETRY
SEA-LEVEL, 59°F DAY, 150 SHP, $\epsilon_R = 0.85$

FIGURE 3-3

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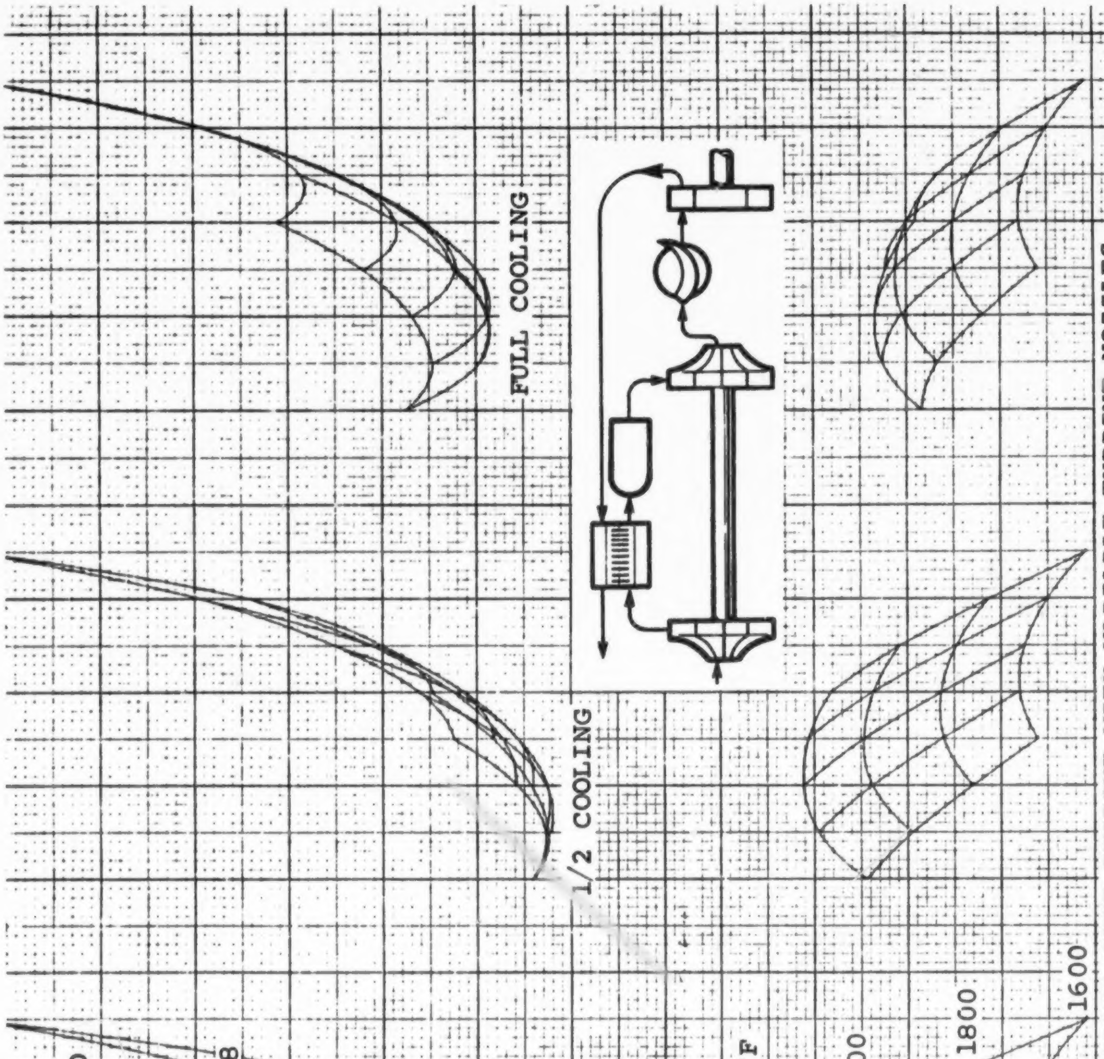

REGENERATED FREE-TURBINE WITH VARIABLE TURBINE NOZZLES
SEA LEVEL, 59°F DAY, 150 SHP, $\epsilon_R = 0.90$

FIGURE 3-4

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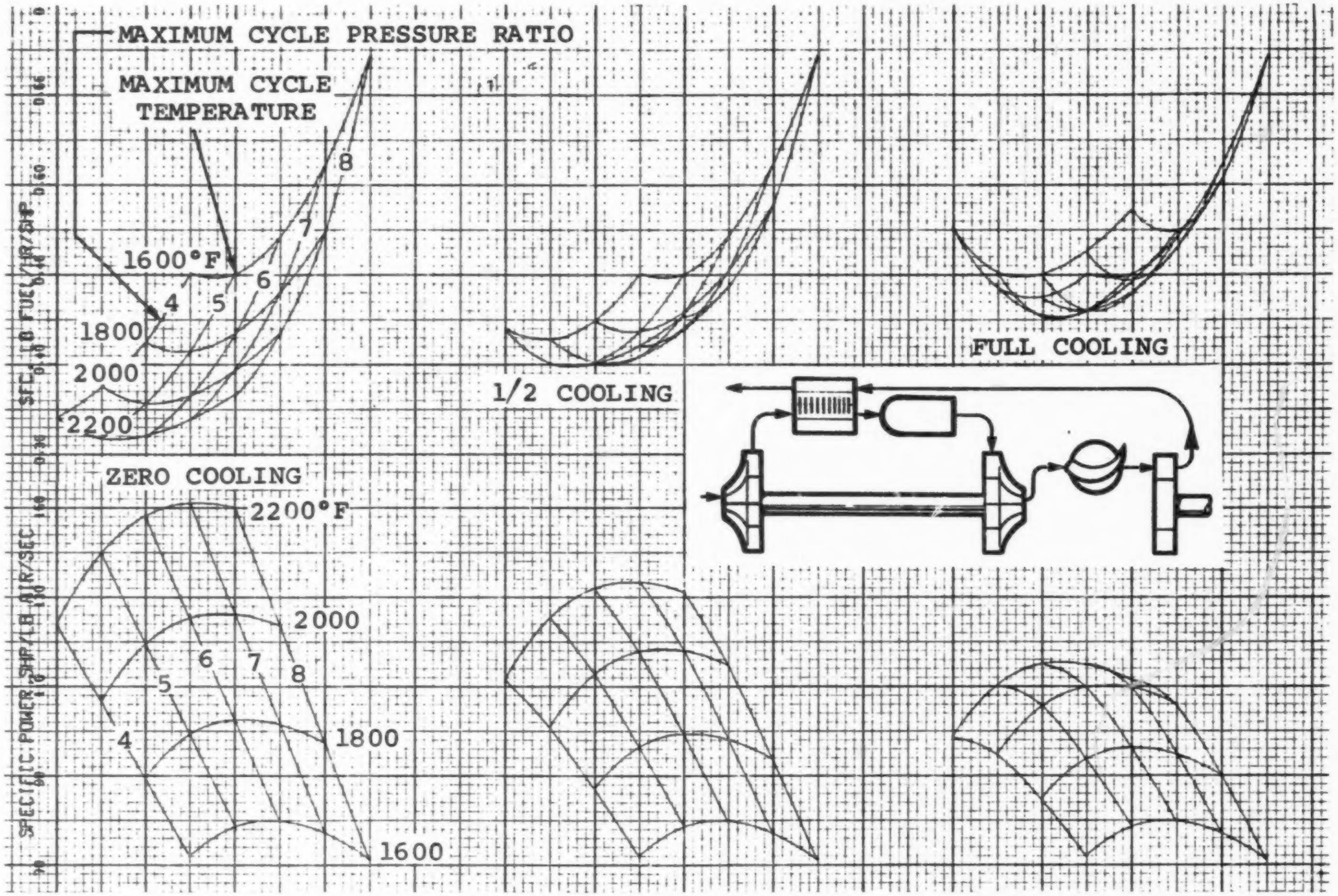

RECUPERATED FREE-TURBINE WITH VARIABLE TURBINE NOZZLES
SEA LEVEL, 59°F DAY, 150 SHP, $\epsilon_R = 0.85$

FIGURE 3-5

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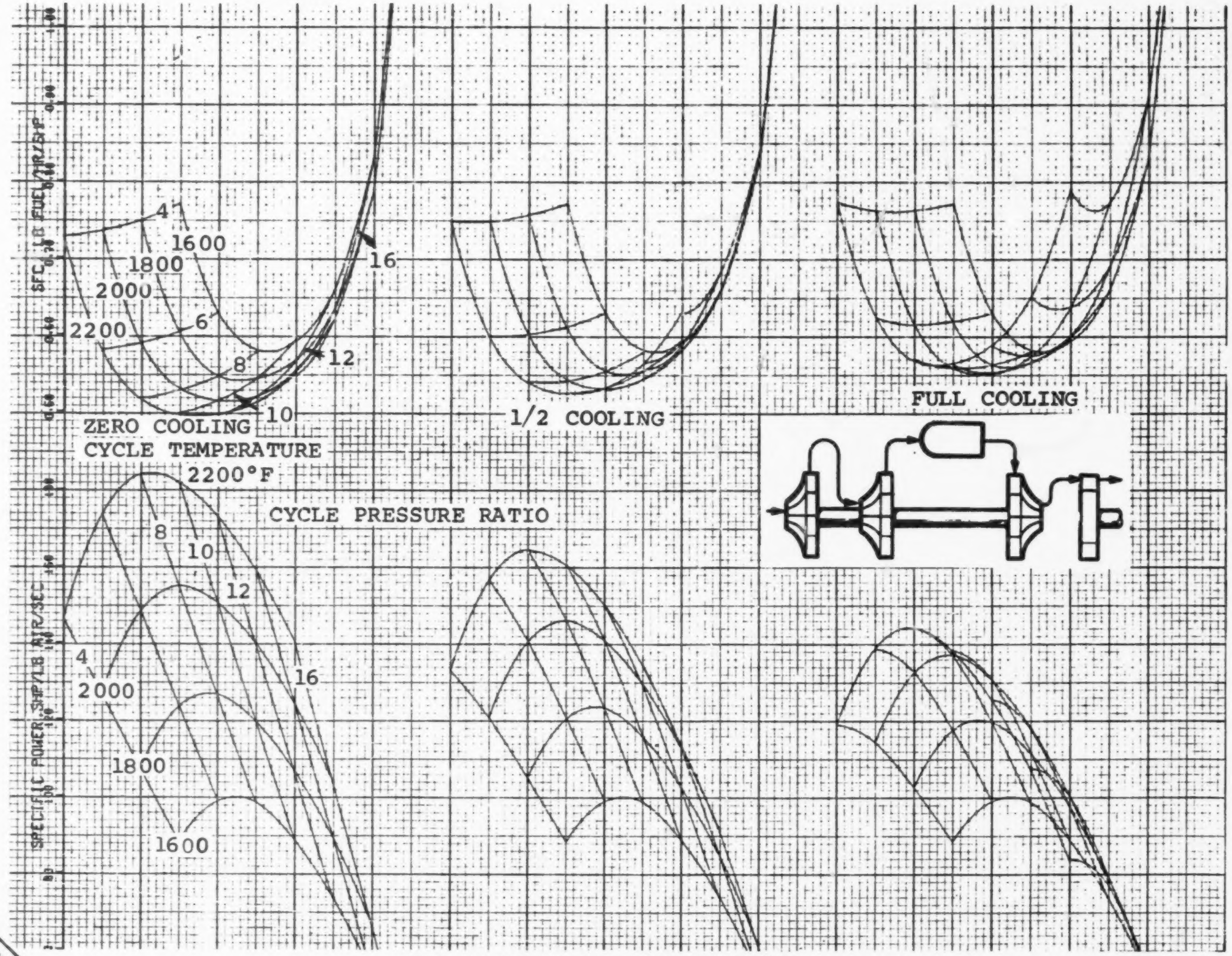

NON-REGENERATED FREE-TURBINE
SEA LEVEL, 59°F DAY, 150 SHP

FIGURE 3-6

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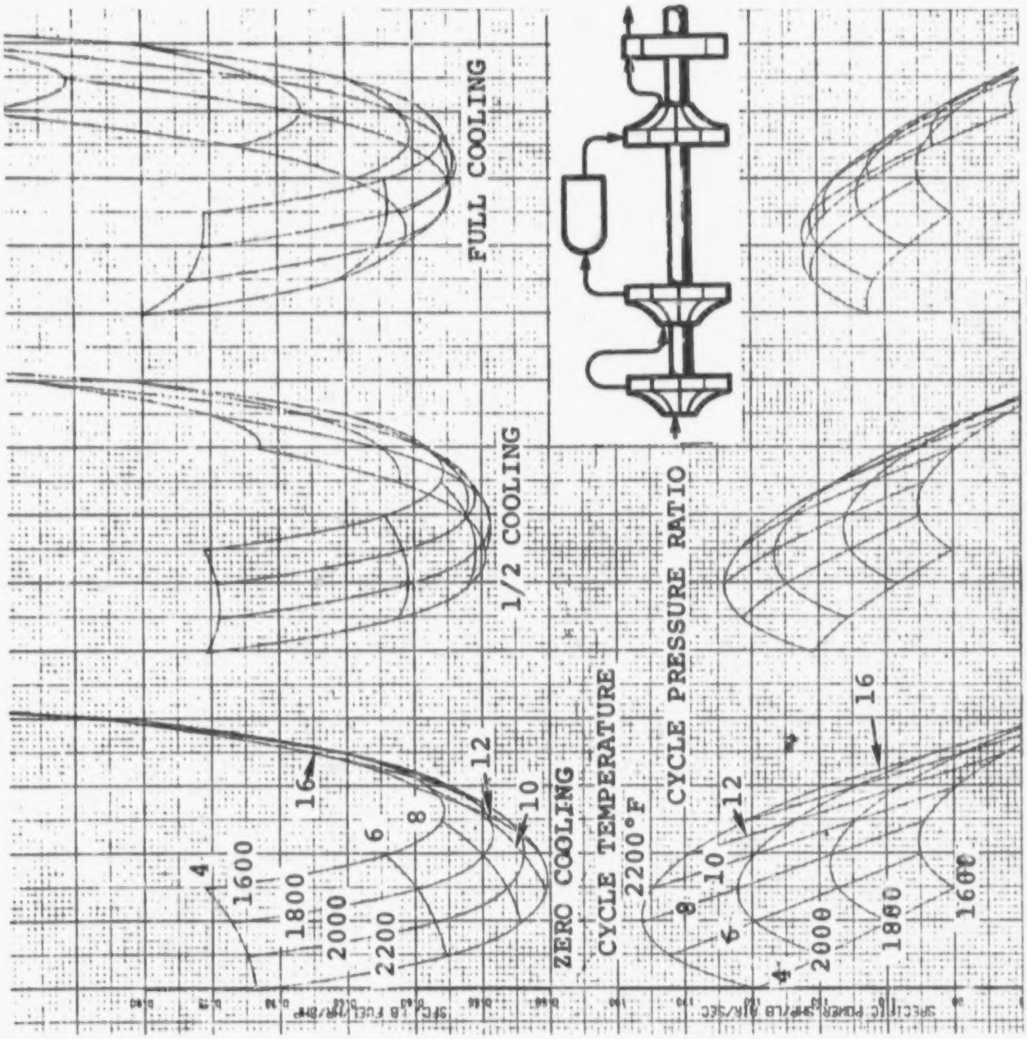

NON-REGENERATED SINGLE-SHAFT
SEA LEVEL, 59°F DAY, 150 SHP

FIGURE 3-7

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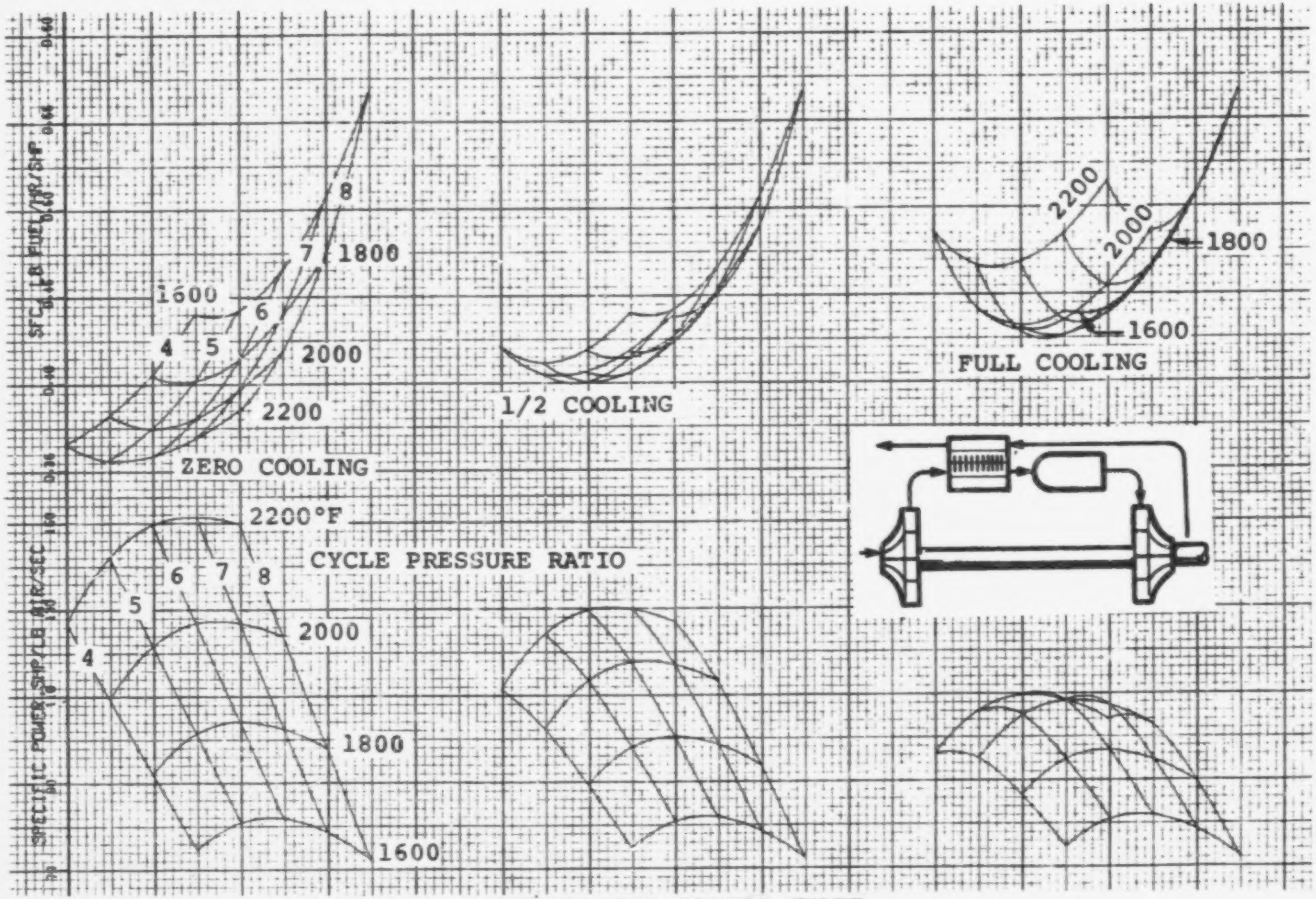

RECUPERATED SINGLE-SHAFT
SEA LEVEL, 59°F DAY, 150 SHP, $\epsilon_R = 0.85$

FIGURE 3-8

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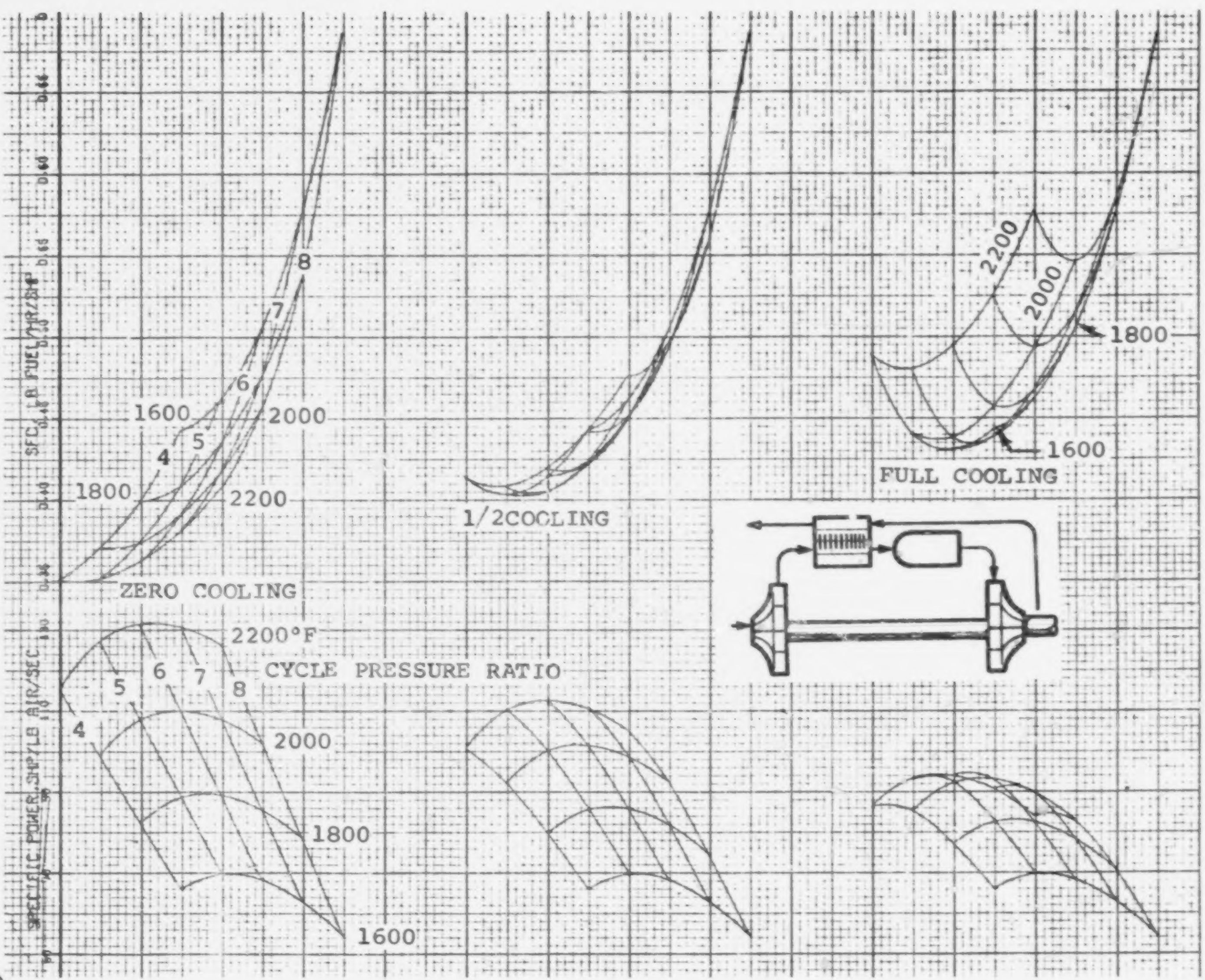
best available copy.



REGENERATED SINGLE-SHAFT
SEA LEVEL, 59°F DAY, 150 SHP, $\epsilon_R = 0.90$

FIGURE 3-9

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SECTION 4

OFF-DESIGN PERFORMANCE ANALYSIS

Since the automobile engine is normally operated at part-load power, it is essential that the performance of a candidate cycle configuration be computed over the entire operating range. Accuracy of the methods employed have been demonstrated at AiResearch through numerous engine development programs in which predicted performance levels were verified by subsequent engine tests.

An existing computer program was the principal tool employed in the off-design analysis. A major program input for an engine configuration was the performance of each component over its operating range. Input concerning the various losses to a given engine are expressed as follows:

(a) Parasitic pressure losses, $\Delta P/P$, were varied as a function of corrected airflow squared $[(W\sqrt{\theta}/\delta)^2]$.

(b) Accessory horsepower losses, including bearings and seals, were varied as a function of shaft speed squared.

(c) Accessory drive gear losses were assigned a value of 2 percent of output shaft horsepower.

(d) Leakage losses were varied directly as the cycle pressure.

The logic flow chart on the following page summarizes basic input information required by this program for each engine. For this study, component off-design characteristics were selected from existing component experimental data and scaled to reflect the component sizes for a 1- to 2-lb/sec through-flow gas turbine engine. Selection criteria

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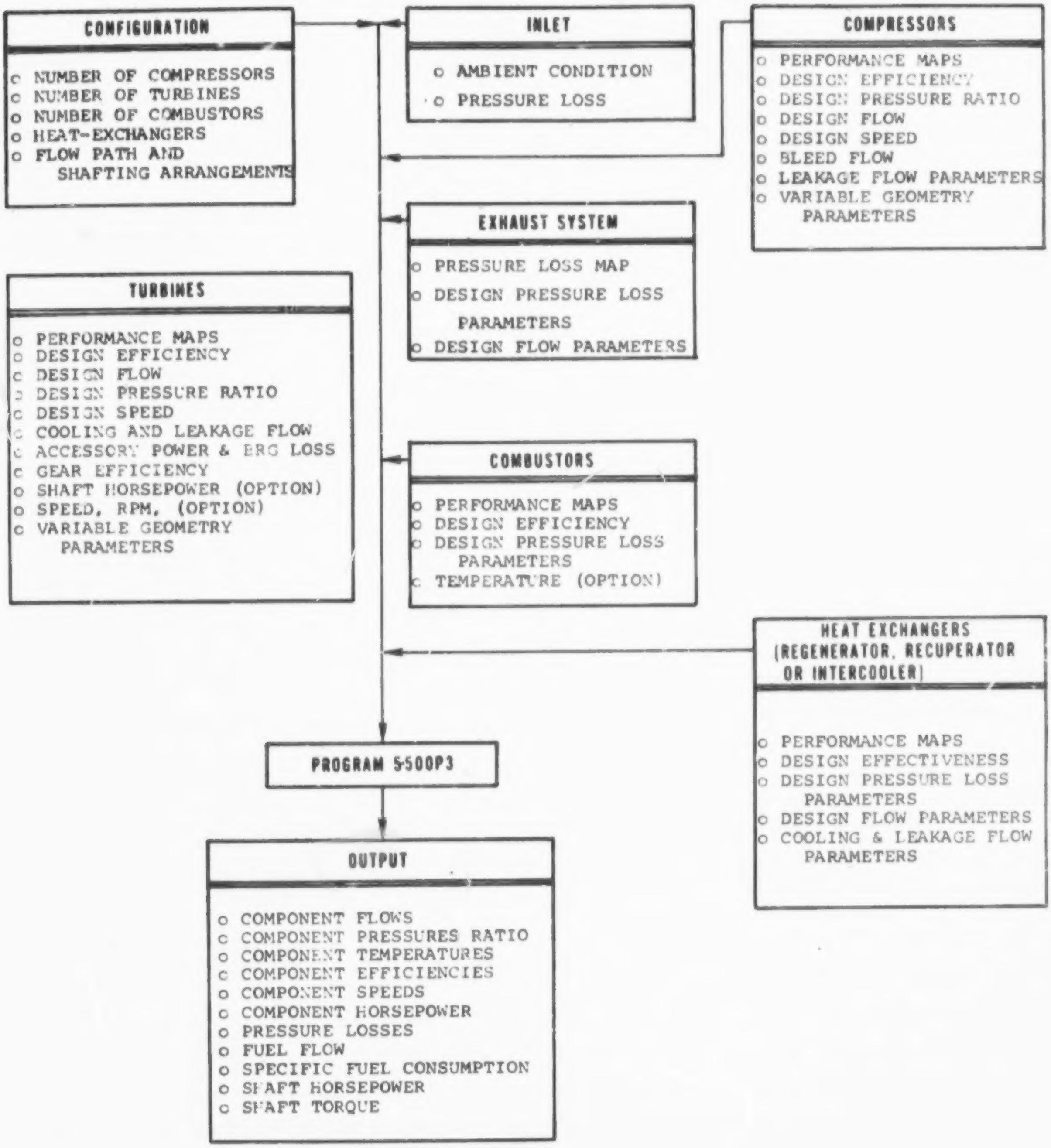
CONFIGURATION
o NUMBER OF COMPRESSORS
o NUMBER OF TURBINES
o NUMBER OF COMBUSTORS
o HEAT-EXCHANGERS
o FLOW PATH AND SHAFTING ARRANGEMENTS
INLET
o AMBIENT CONDITION
o PRESSURE LOSS
COMPRESSORS
o PERFORMANCE MAPS
o DESIGN EFFICIENCY
o DESIGN PRESSURE RATIO
o DESIGN FLOW
o DESIGN SPEED
o BLEED FLOW
o LEAKAGE FLOW PARAMETERS
o VARIABLE GEOMETRY PARAMETERS
EXHAUST SYSTEM
o PRESSURE LOSS MAP
o DESIGN PRESSURE LOSS PARAMETERS
o DESIGN FLOW PARAMETERS
TURBINES
o PERFORMANCE MAPS
o DESIGN EFFICIENCY
o DESIGN FLOW
o DESIGN PRESSURE RATIO
o DESIGN SPEED
o COOLING AND LEAKAGE FLOW
o ACCESSORY POWER & BRG LOSS
o GEAR EFFICIENCY
o SHAFT HORSEPOWER (OPTION)
o SPEED, RPM, (OPTION)
o VARIABLE GEOMETRY PARAMETERS
COMBUSTORS
o PERFORMANCE MAPS
o DESIGN EFFICIENCY
o DESIGN PRESSURE LOSS PARAMETERS
o TEMPERATURE (OPTION)
HEAT EXCHANGERS (REGENERATOR, RECUPERATOR OR INTERCOOLER)
o PERFORMANCE MAPS
o DESIGN EFFECTIVENESS
o DESIGN PRESSURE LOSS PARAMETERS
o DESIGN FLOW PARAMETERS
o COOLING & LEAKAGE FLOW PARAMETERS
PROGRAM 5-500P3
OUTPUT
o COMPONENT FLOWS
o COMPONENT PRESSURES RATIO
o COMPONENT TEMPERATURES
o COMPONENT EFFICIENCIES
o COMPONENT SPEEDS
o COMPONENT HORSEPOWER
o PRESSURE LOSSES
o FUEL FLOW
o SPECIFIC FUEL CONSUMPTION
o SHAFT HORSEPOWER
o SHAFT TORQUE

OFF-DESIGN COMPUTER PROGRAM

for the component performance maps included pressure ratio and Mach number levels comparable to the candidate cycle components. These criteria assured off-design characteristics equivalent to the scaled experimental components. Therefore, by specifying a range of input parameters, engine performance maps for the entire operating range were generated. At each match-point on the engine map, the following data were available for input to the mission analysis computer program described in Section 6 of this report:

(a) Shaft horsepower
(b) Output shaft speed
(c) Fuel flow rate
(d) Combustor inlet temperature
(e) Combustor exit temperature
(f) Combustor inlet and exit pressure
(g) Combustor efficiency
(h) Fuel-air ratio
(i) Airflow rate
(j) Gasifier speed
(k) Transient performance (for free-turbine cycle)

Only the 1975 level of technology was used in the off-design performance analyses. The maximum power for all candidate cycles, without power augmentation, was assigned 150 shaft horsepower (shp) on a sea-level, standard day (59°F, 14.696 psia). The engines selected for power augmentation were assigned 108 shp. Therefore, all component efficiencies were based on components sized for this power class. However, the mission analysis computer program has the capability of scaling engine performance without component efficiency modifications, to match the power requirements of a given engine and transmission configuration.

Initial studies showed that vehicle acceleration requirements established the maximum power level of the engine. However, since a large percentage of engine operating time is at a low power level, a compromise was sought that would allow lower levels of design-point horsepower, thereby improving part-load specific fuel consumption. Reduction of design-point power can be achieved by supplying some means of power augmentation sufficient to comply with the transient vehicle acceleration requirements. Consequently, to improve low-power specific fuel consumption and reduce engine size, different methods of providing power boost were considered and are discussed in the following paragraphs.

4.1 REHEAT

The amount of reheat (power boost) that can be used between turbine stages is a function of the maximum cycle temperature, cycle pressure ratio, and the work split between turbine stages. Calculations were made to determine the compressor and turbine size reduction that could be achieved with reheat using a 150-hp, low-pressure-ratio (4.6:1), regenerative, free-turbine engine as the basic cycle. Because of the relatively high Mach number (0.45 to 0.5) at the gasifier turbine exit, the interstage burner pressure loss, $\Delta P/P$, was estimated as 7 percent. Respective cycle flow or engine size reductions of approximately 4 and 2 percent were computed, using a maximum cycle temperature of 1850°F and a resultant 330°F of reheat. Also, calculations showed that maximum flow and size reductions of approximately 20 and 10 percent, respectively, could be obtained with reheat for a 12:1 pressure-ratio, free-turbine engine having a maximum cycle temperature of 1900°F. Because of the large burner volume, increased complexity, and reduction of turbine life, the reheat burner does not offer a high potential for a low-cost automobile engine.

4.2 AXIAL SUPERCHARGING

This method of power employs an axial compressor stage upstream from the centrifugal compressor to boost cycle pressure, when maximum power is required. For lower power operation, the axial stage is disconnected from the engine.

The general problem concerning the single-stage axial compressor in the development of maximum acceleration power is simply stated: There must be an effective means to remove the axial stage from the system for operation at low power to minimize inlet pressure loss. Three alternative arrangements, all incorporating a declutching feature, might be considered:

(a) A conventional straight-through flow path with the axial rotor windmilling

(b) A conventional straight-through flow path with the axial rotor feathered, as in variable pitch propeller applications

(c) A bypass flow path to the centrifugal compressor that would probably require two-position doors

Nonsupercharging losses in the first two arrangements can be as large as one through-flow velocity head or nearly 10 percent of the inlet pressure. Because inlet losses, for both connected and disconnected operating conditions, indicate a potential for being less, the third arrangement shown schematically in Figure 4-1 was explored in more detail. Table 4-1 summarizes the results of calculations with related aerodynamic properties. An axial stage with a total pressure ratio of 1.6 and 85-percent efficiency was selected.

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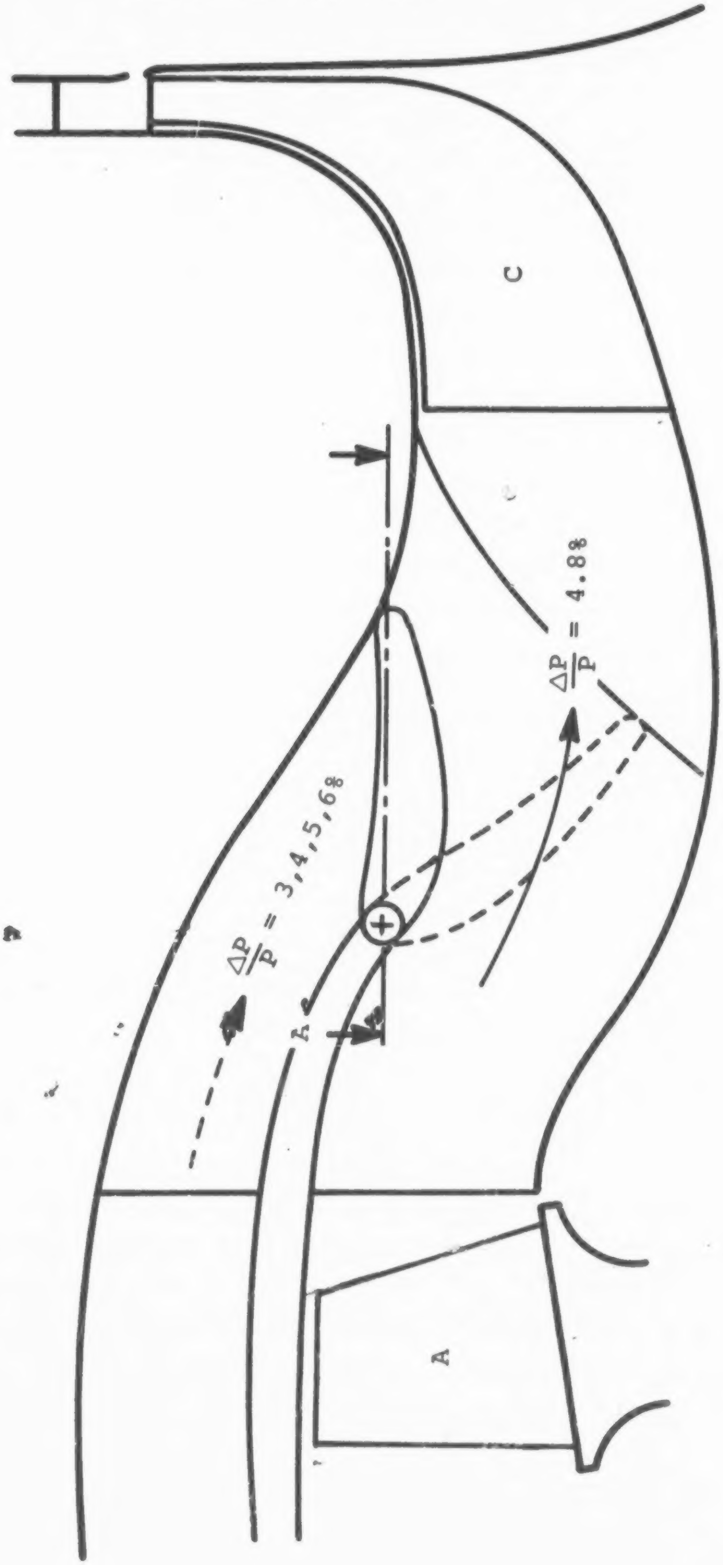

C
$\frac{\Delta P}{P} = 4.8\%$
$\frac{\Delta P}{P} = 3,4,5,6\%$
A

AXIAL STAGE SUPERCHARGING SCHEMATIC

FIGURE 4-1

TABLE 4-1

COMPARISON OF COMPONENT SIZES -
AXIAL SUPERCHARGED VERSUS NON-SUPERCHARGED ENGINE

Engine	Operation Mode at Maximum shp		Pressure Loss Upstream from Centrifugal Compressor, % of Ambient Pressure	Corrected Flow Into Centrifugal Compressor at Maximum shp, lb/sec	Diameter Reduction for Same Maximum shp, %
	Non-supercharged	Supercharged			
Non-supercharged	136.5	--	1.5	1.332**	--
	141.4	--	↓	1.380**	--
	146.8	--		1.433**	--
	108	--		1.054	--
Supercharged	108	136.5*	3.0	1.092	9.5
	↓	141.4*	4.5	1.130	↓
		146.8*	6.0	1.171	

*Assuming a 4.8-percent loss in axial discharge pressure between the axial and centrifugal stage

**Scaled from value for 108-hp engine (1.054)

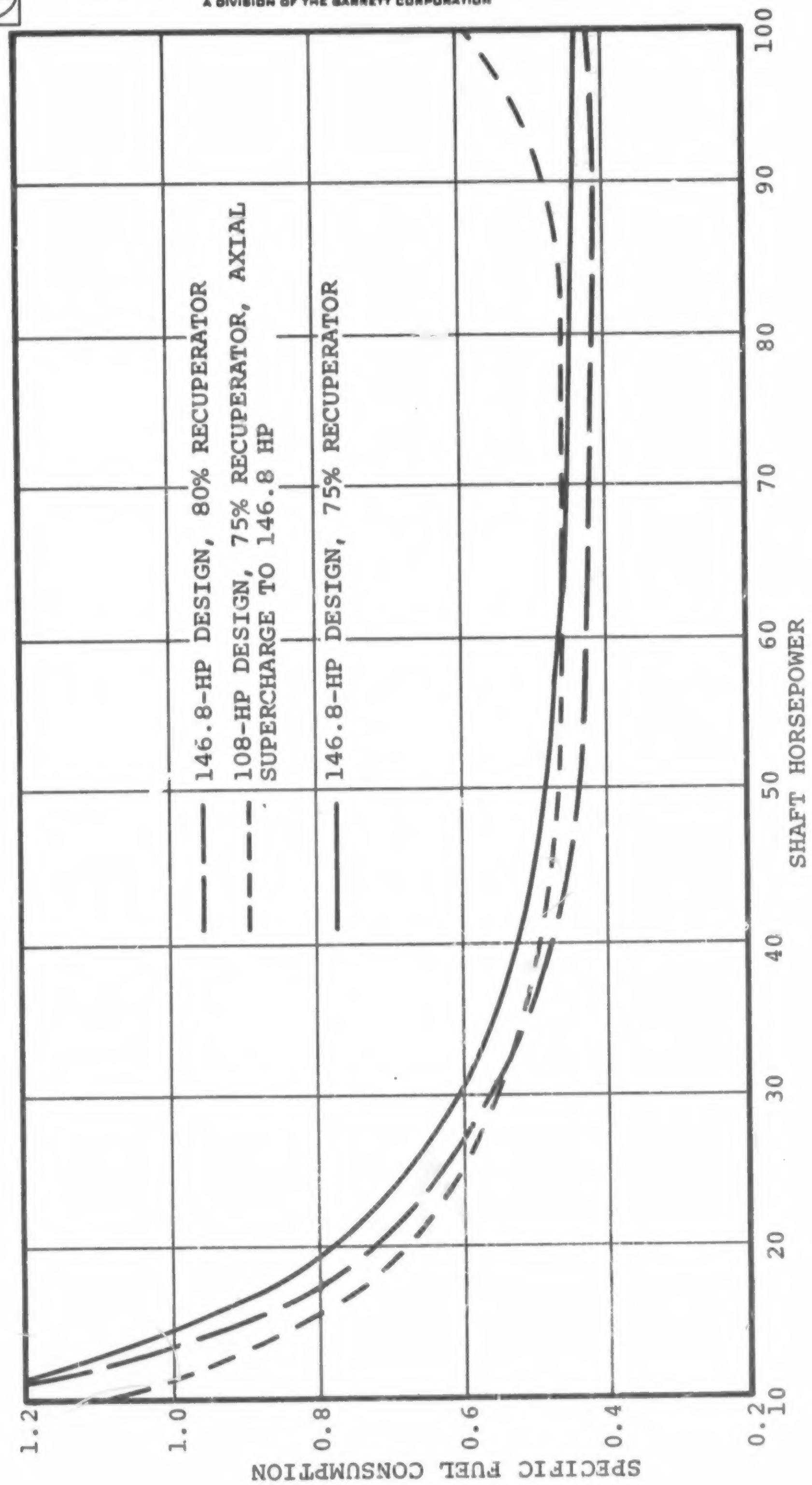
Bypass inlet and interstage pressure losses are variables in the calculation. Base inlet loss for a typical 108-hp engine is 1.5 percent of inlet pressure. When the inlet loss is increased, 108 hp can be developed by increasing the centrifugal compressor corrected flow. To maintain the same stall margin, the turbine flow function must increase in direct proportion to the compressor flow. Table 4-1 also indicates the magnitude of the changes for inlet loss values of 3, 4.5, and 6 percent of inlet pressure. The loss between the supercharging axial and the centrifugal compressor was assumed to be 4.8 percent.

If the base engine was scaled to achieve the same power as axial supercharging with a 4.8-percent interstage loss, the centrifugal compressor flow would increase. Comparing this flow to the centrifugal compressor flow of the axial supercharged engine shows the size reduction that accrues from axial supercharge. Irrespective of the bypass inlet pressure loss, the reduction in centrifugal compressor diameter (in changing from a centrifugal compressor sized for acceleration power to a supercharged centrifugal for the same acceleration power) is only 9.5 percent.

Better appreciation for the merits of axial supercharging may be gained from Figure 4-2. For a 75-percent effectiveness recuperator, the axial supercharged design offers good advantage over the larger centrifugal design in the power range below approximately 60 hp, which may partially overcome the additional complexities of axial supercharging. However, improving the recuperator effectiveness to 80 percent reduces this advantage except in the very low power range.

Axial supercharging could be more attractive, depending upon additional work (beyond the scope of this study) in the following problem areas:

(a) Innovation in geometric arrangement to reduce bypass inlet and transition pressure losses



RESULTS OF AXIAL SUPERCHARGING, 85°F DAY

FIGURE 4-2

(b) Higher supercharging pressure ratios with reasonable efficiency

(c) Better match between typical through-flow axial and centrifugal Mach numbers to reduce transition pressure loss

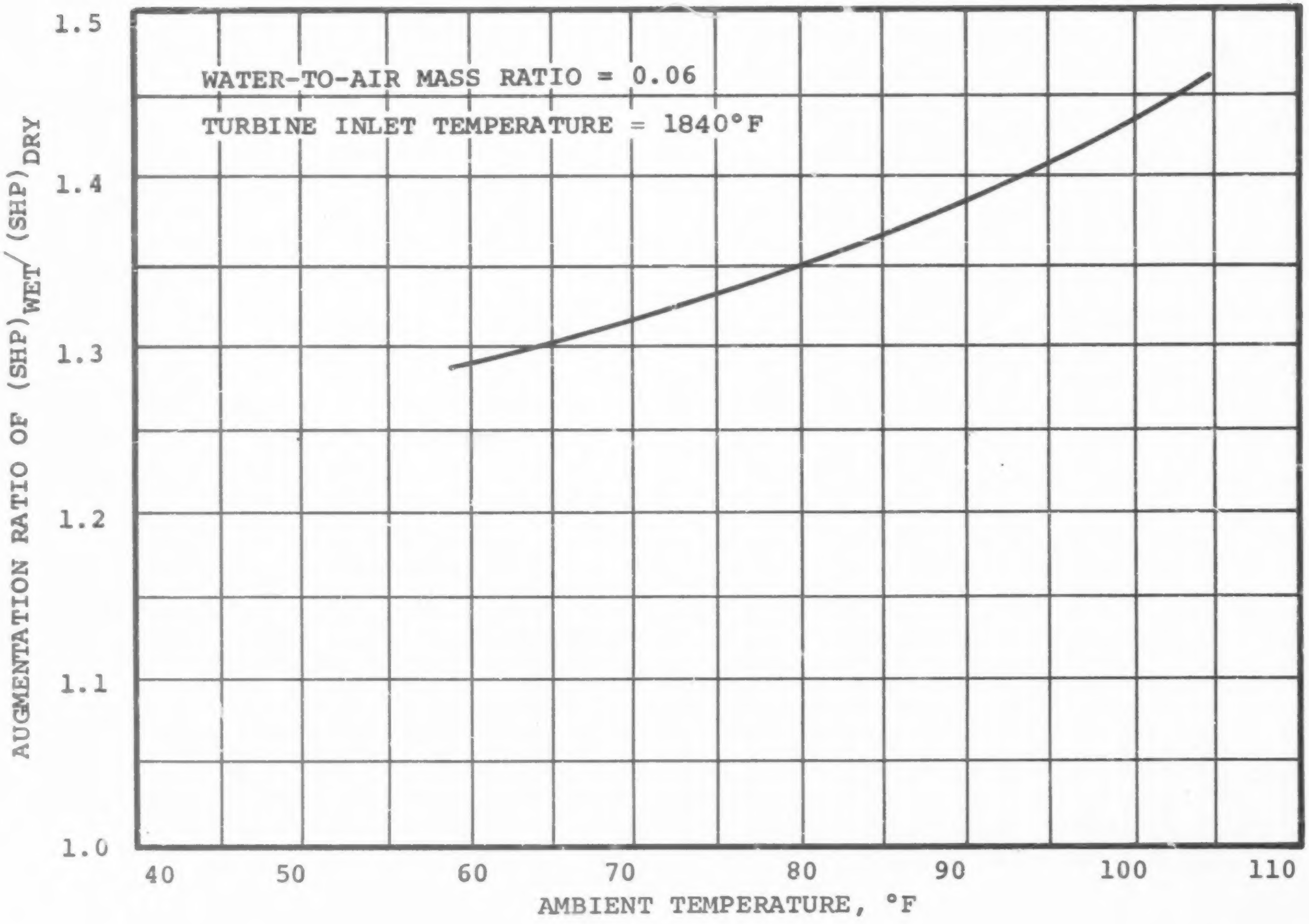
(d) Clutch mechanical design, response time, and life

4.3 WATER INJECTION

The power of a gas turbine engine may be increased by injecting a suitable liquid into the engine airflow path. Injection may be accomplished through spray nozzles at the compressor inlet, at various interstage points within the compressor, at the combustor inlet, or a combination of these points. For this study, only water injected at the compressor inlet was considered, since this point in the cycle provides maximum power boost. The theoretical and experimental results of water and water-methanol mixtures, injected into the inlet of the Garrett/AiResearch T-76 turboprop engine, were reported in 1970[6]. These results showed that a power-augmentation ratio $[(shp)_{wet}/(shp)_{dry}]$ of 1.4 could be achieved with a water-to-air mass ratio of 0.06, when the engine is operating on a 100°F day with constant turbine exit temperature (Figure 4-3). The ratio of specific fuel consumption $[(sfc)_{wet}/(sfc)_{dry}]$ was essentially constant.

No detailed calculations were made of engine performance with water injection. However, based on the above results, a maximum augmentation ratio of 1.3 was used for this study.

Water injection would be used only as a power-boost medium for acceleration and similar maximum power conditions and would require only a small reservoir (Appendix 2). The engine would be sized so that power boost would be required only for ambient temperatures in excess of 60°F.



POWER AUGMENTATION WITH CONSTANT WATER-TO-AIR MASS RATIO

FIGURE 4-3

Multiple-spool cycles were considered but not specifically studied in detail for this application. Considerable work has been done previously on vehicular application of the gas turbine engine, results of which indicate that multiple-spool cycles have a number of favorable features:

(a) The variable-speed turbocharger produces results similar to that of adding variable geometry to the 1- or 1-1/2-spool cycles and also provides excellent part-load fuel economy.

(b) Turbine inlet temperature varies with load, lengthening stress-rupture, and oxidation-life at part-load.

(c) Two compressor stages are available for sharing the cycle pressure ratio, with potentially higher overall compressor efficiencies than obtainable in single-stage compressors of the same overall pressure ratio.

Undesirable characteristics of these same cycles are:

(a) High-inertia, variable-speed turbocharger (low-pressure spool) must be accelerated to rated speed to produce rated power. Thus, engine response to load-demand is extremely slow compared to the "instantaneous" power capability of spark ignition (SI) engines and single-shaft, gas turbine engines.

(b) Rematching the pressure-ratio split between spools for improved response inflicts a significant penalty on part-load fuel economy.

(c) Power output and fuel economy are highly influenced by variations in accessory power loading on the high-pressure spool and by small variations in component characteristics and "build" techniques that normally are within acceptable production tolerances for simple cycles.

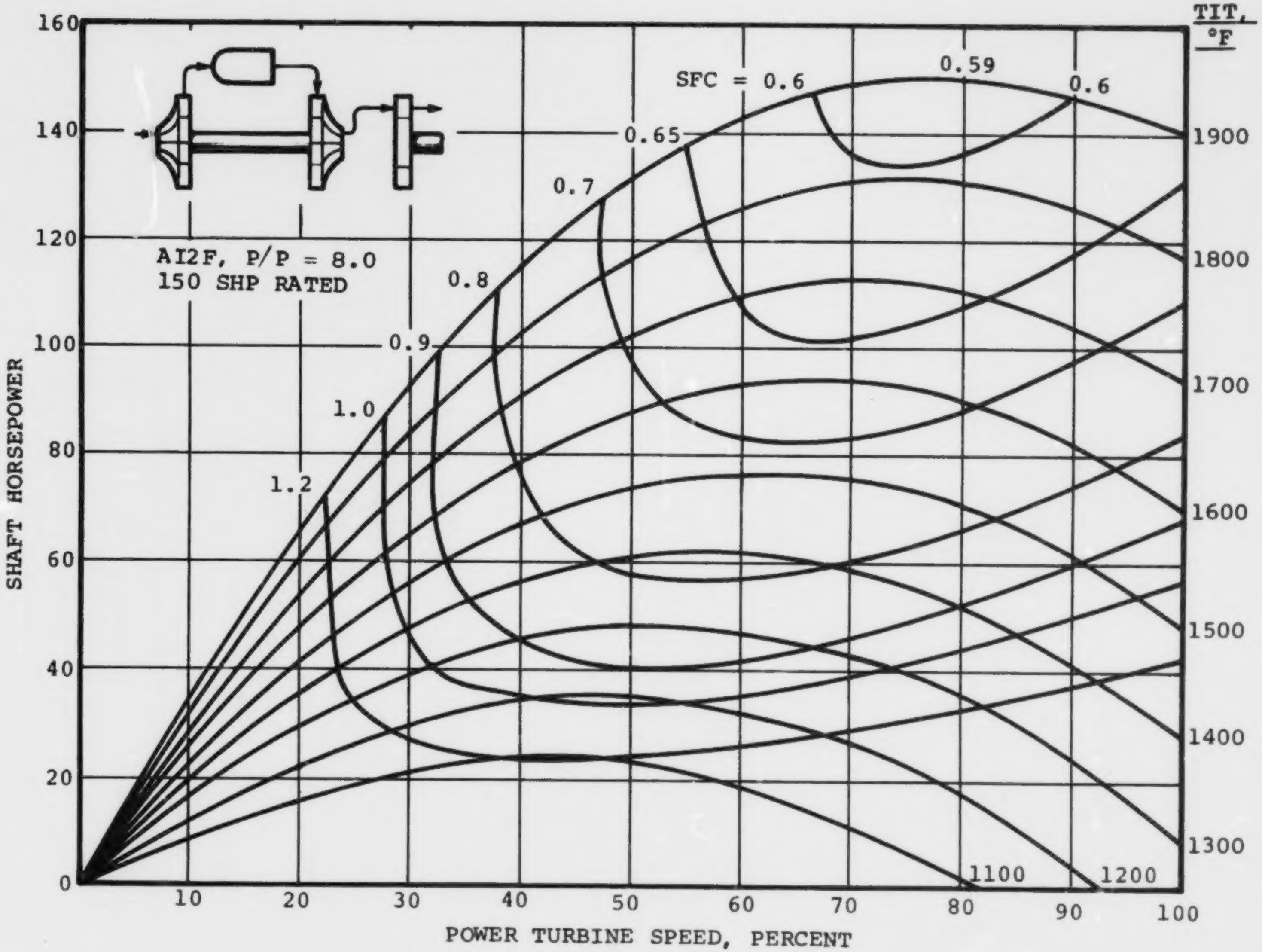
(d) Cycle packaging arrangements dictated by available automobile engine envelopes may result in excessive interstage cycle losses which could negate some of the attractive part-load performance characteristics.

(e) Engine braking is not available without introducing cycle complexities, such as interspool clutches and compressor bleed valves with associated controls, shafts, and gears.

(f) Power-lapse rate with ambient temperature is excessively large and requires oversizing to achieve the necessary power output on hot days.

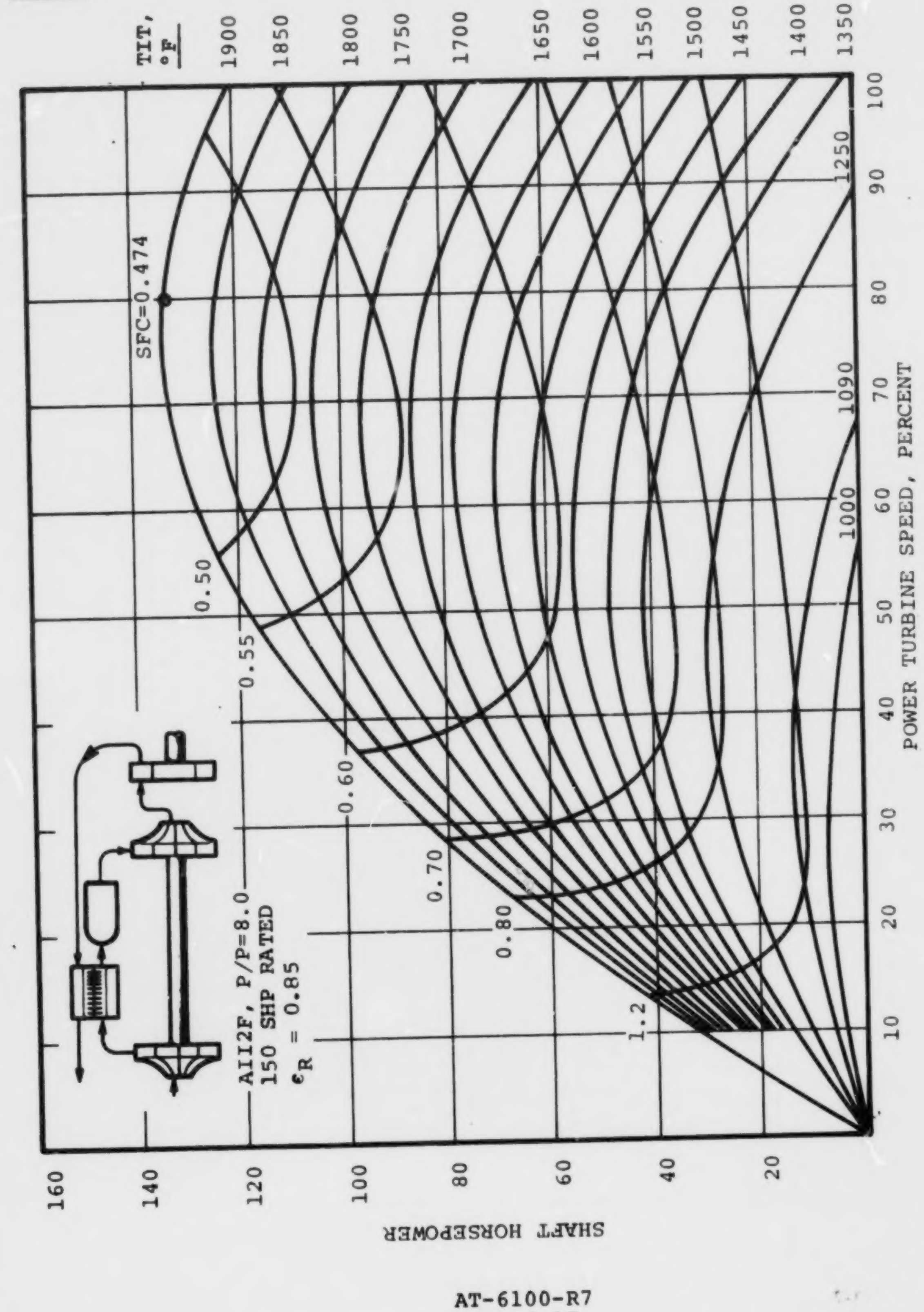
Summation of the preceding factors implies excessive weight, cost, and complexity. The disadvantages of multiple-spool cycles seem to far out-weigh the advantages, thus making them unattractive for the automobile application.

Figures 4-4 through 4-16 show typical off-design engine performance for each of the engine configurations listed in Table 4-2. These maps are a result of an iterative process involving the calculation of off-design engine performance for each cycle listed, starting with the simplest. This output is then used as input to the mission analysis program to determine whether or not the engine/vehicle performance goals were achieved. If the automobile system fuel economy did not prove satisfactory, various cycle modifiers were added to improve engine cycle efficiency, and the iterative process was repeated. Cycle modifiers investigated include:



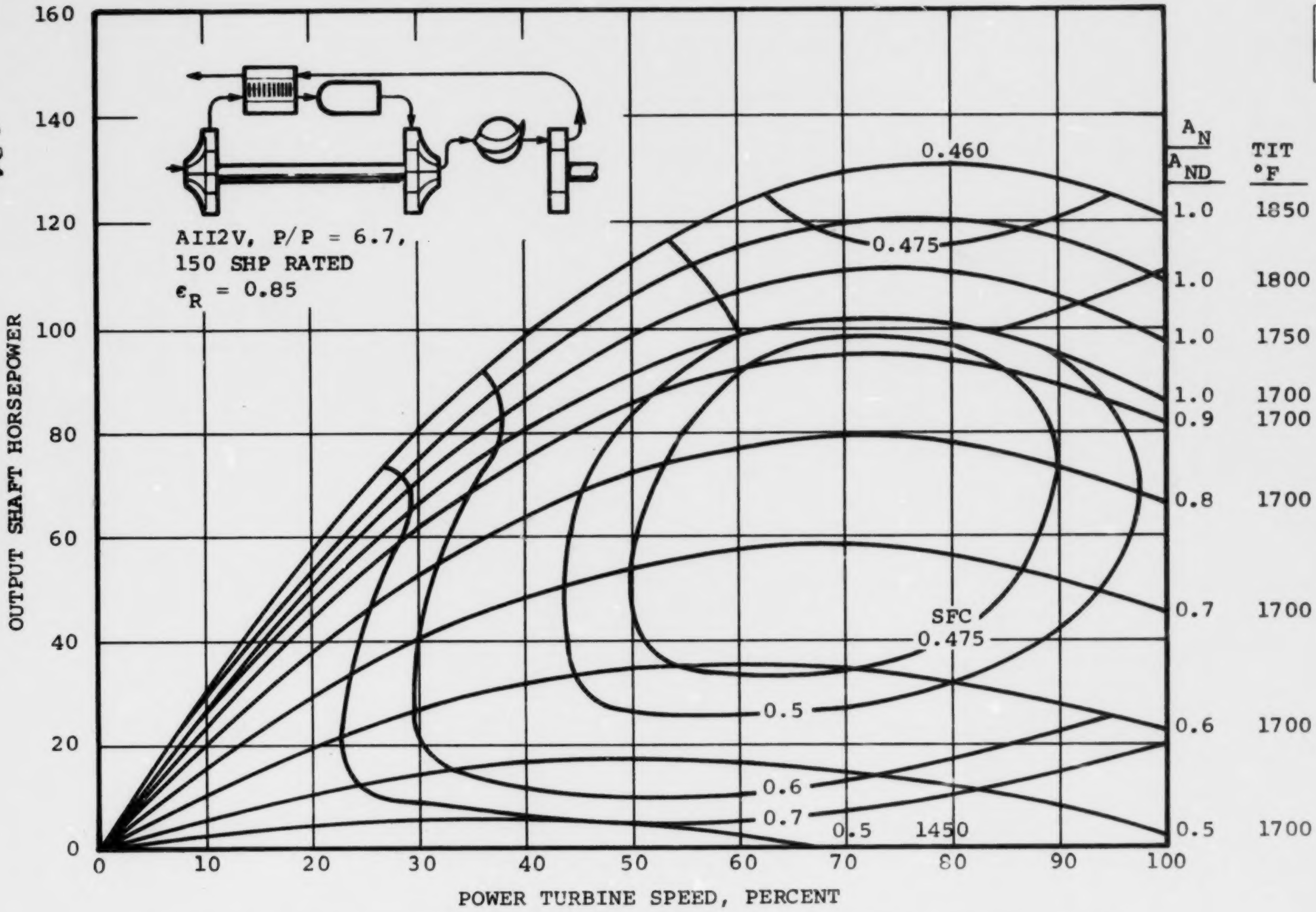
FREE-TURBINE CYCLE, 59°F DAY

FIGURE 4-4



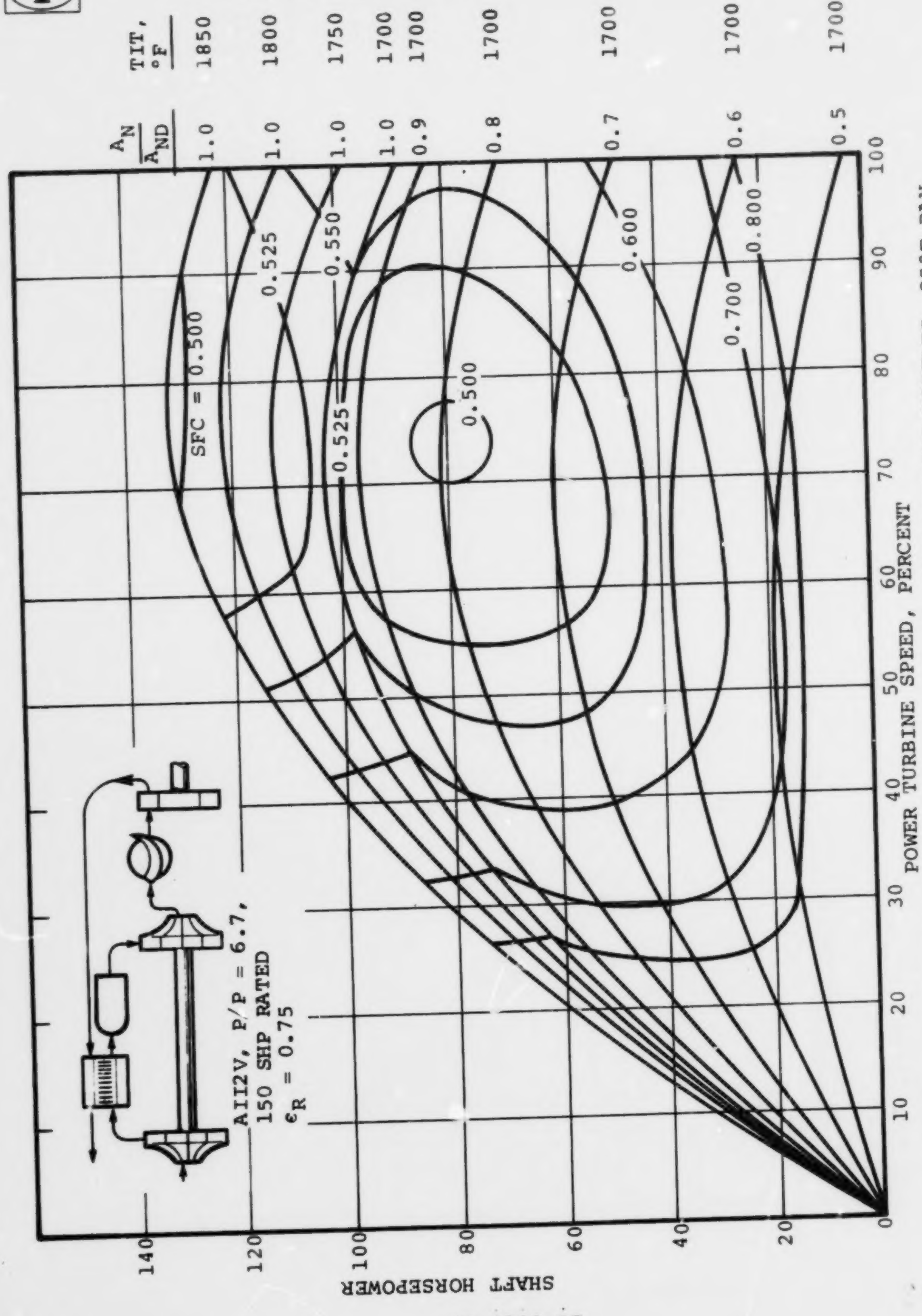
RECUPERATED FREE-TURBINE CYCLE, 85°F DAY

FIGURE 4-5



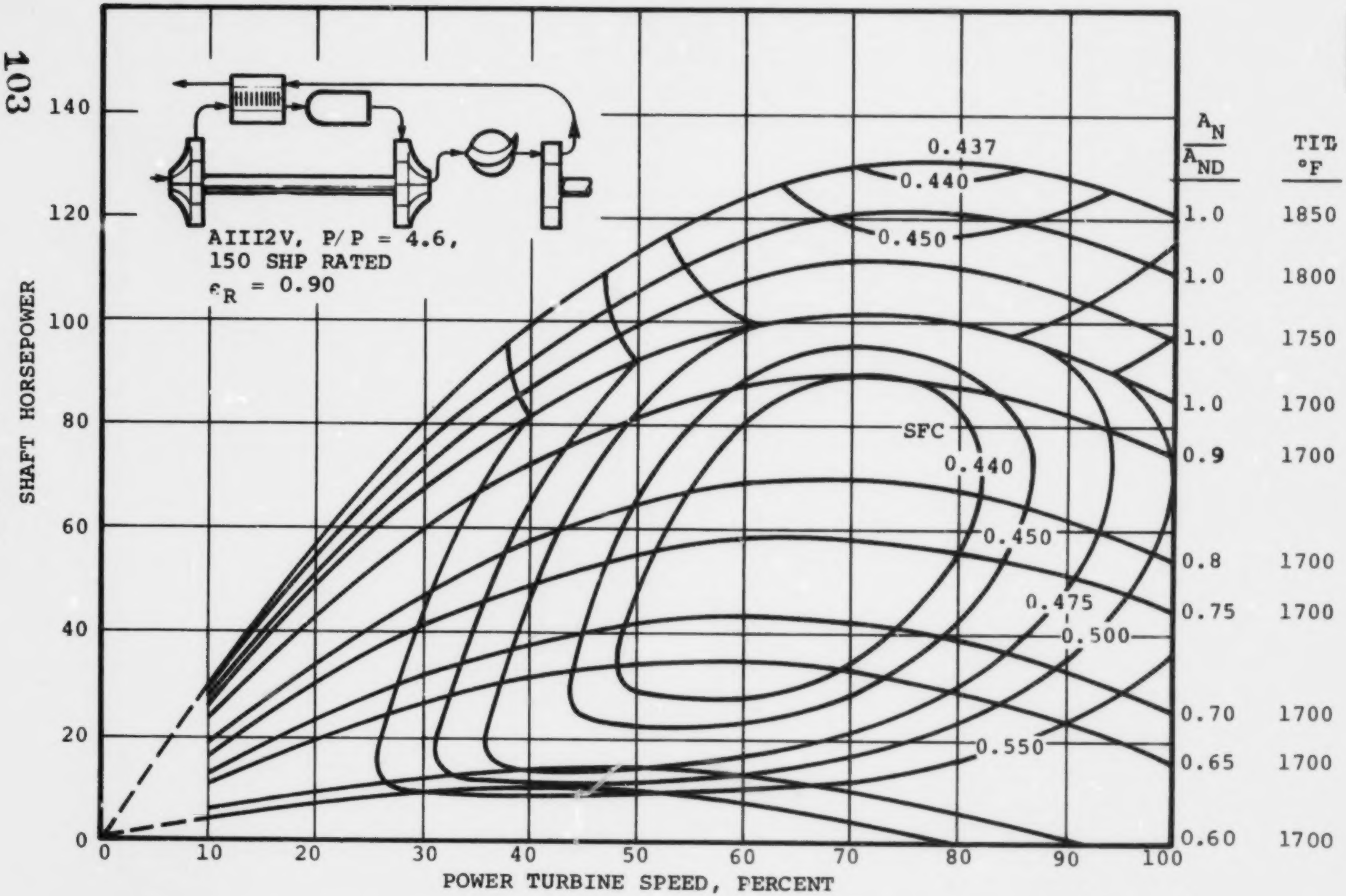
RECUPERATED FREE-TURBINE CYCLE WITH VARIABLE POWER TURBINE NOZZLES, 85°F DAY

FIGURE 4-6



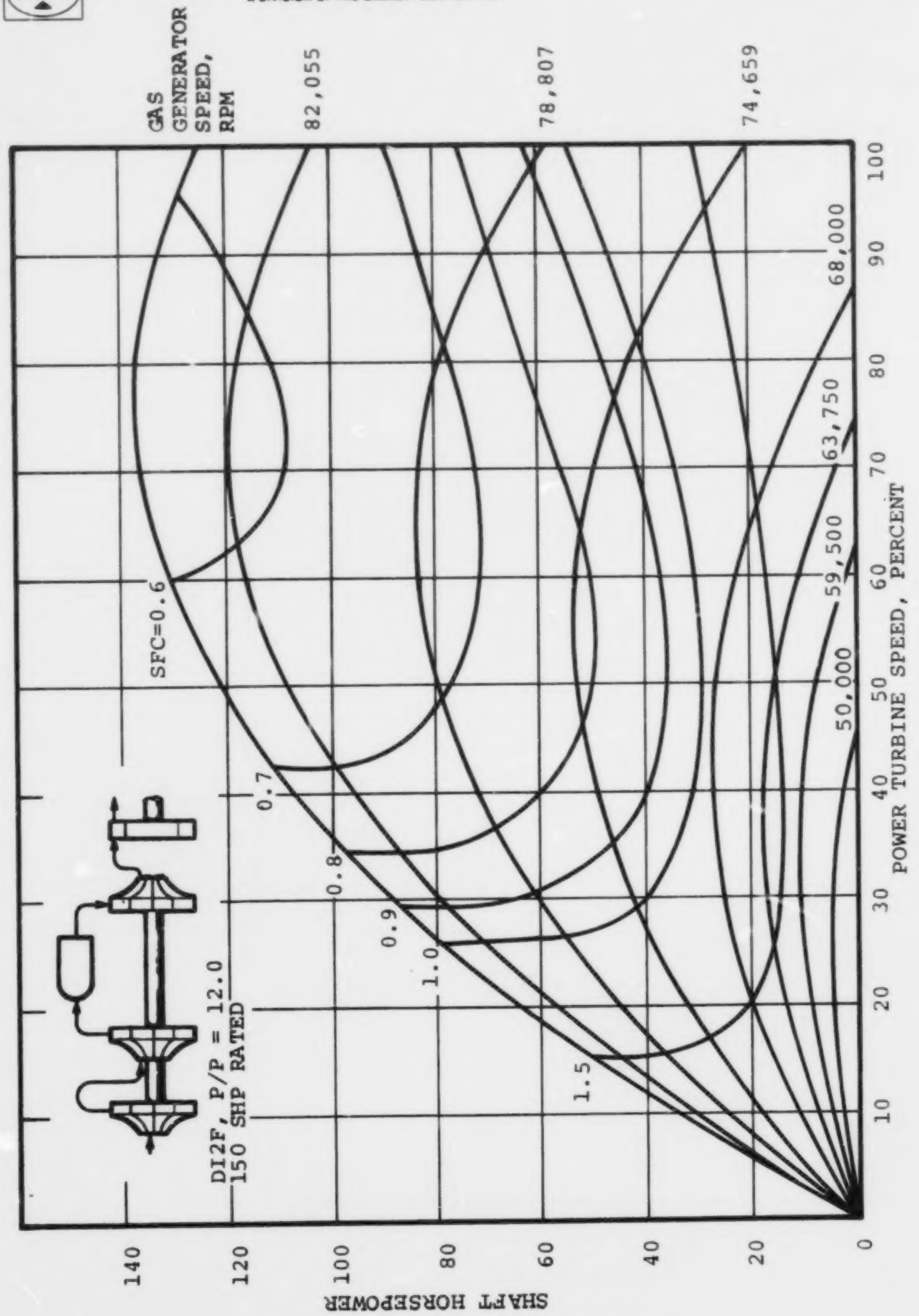
RECUPERATED FREE-TURBINE CYCLE WITH VARIABLE POWER TURBINE NOZZLES, 85°F DAY

FIGURE 4-7



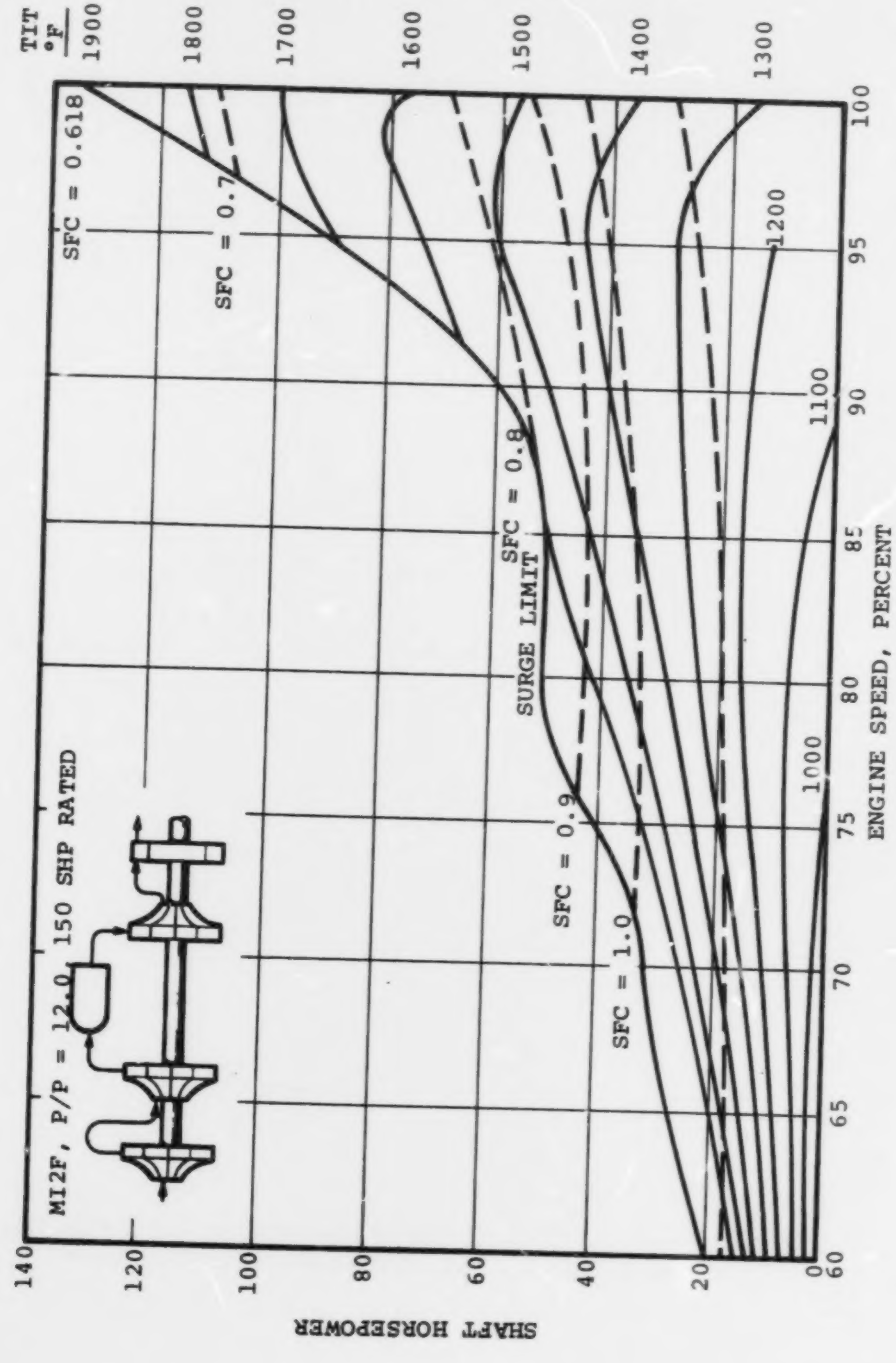
REGENERATED FREE-TURBINE CYCLE WITH VARIABLE POWER TURBINE NOZZLES, 95°F DAY

FIGURE 4-8



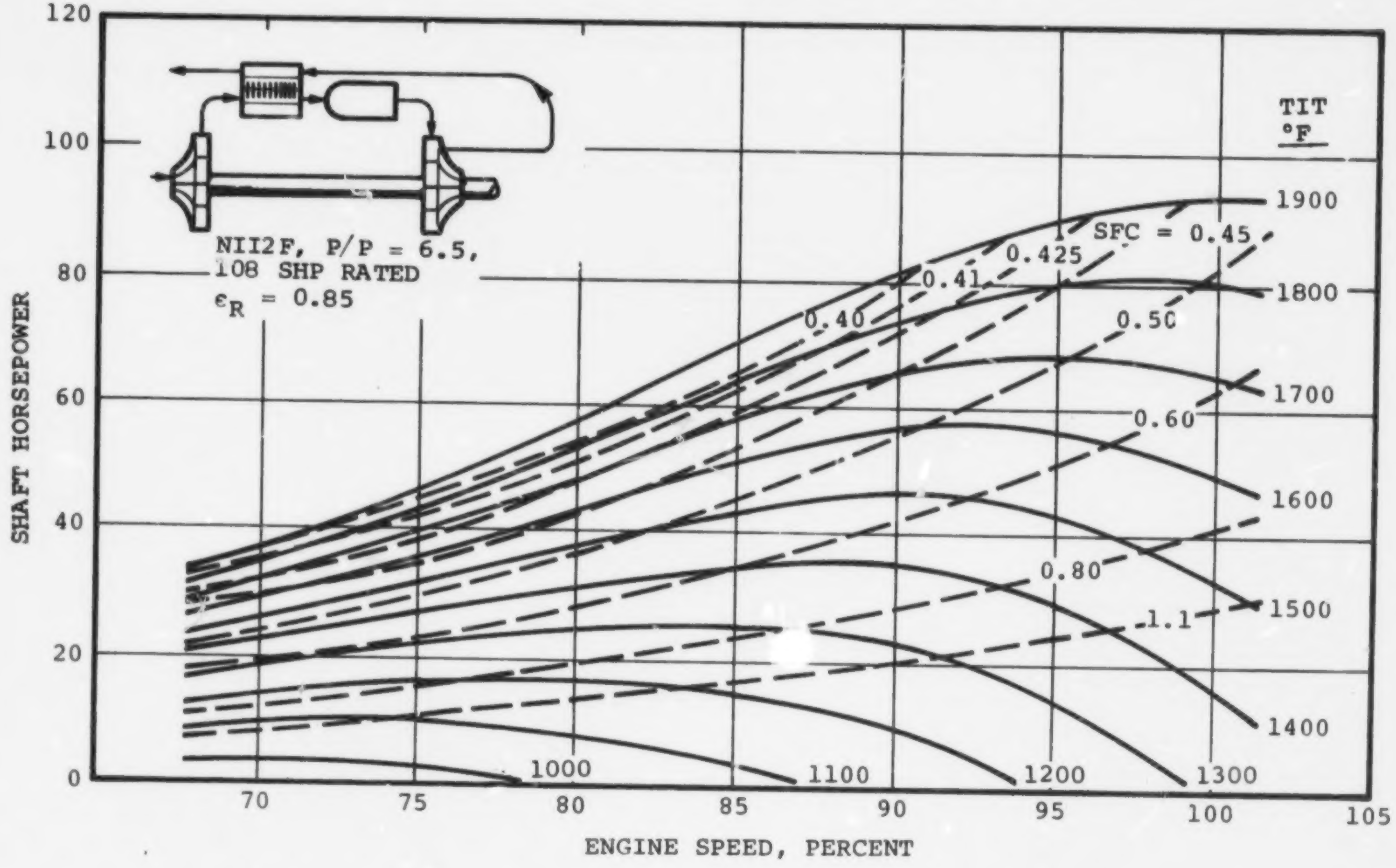
FREE-TURBINE CYCLE, 85°F DAY

FIGURE 4-9



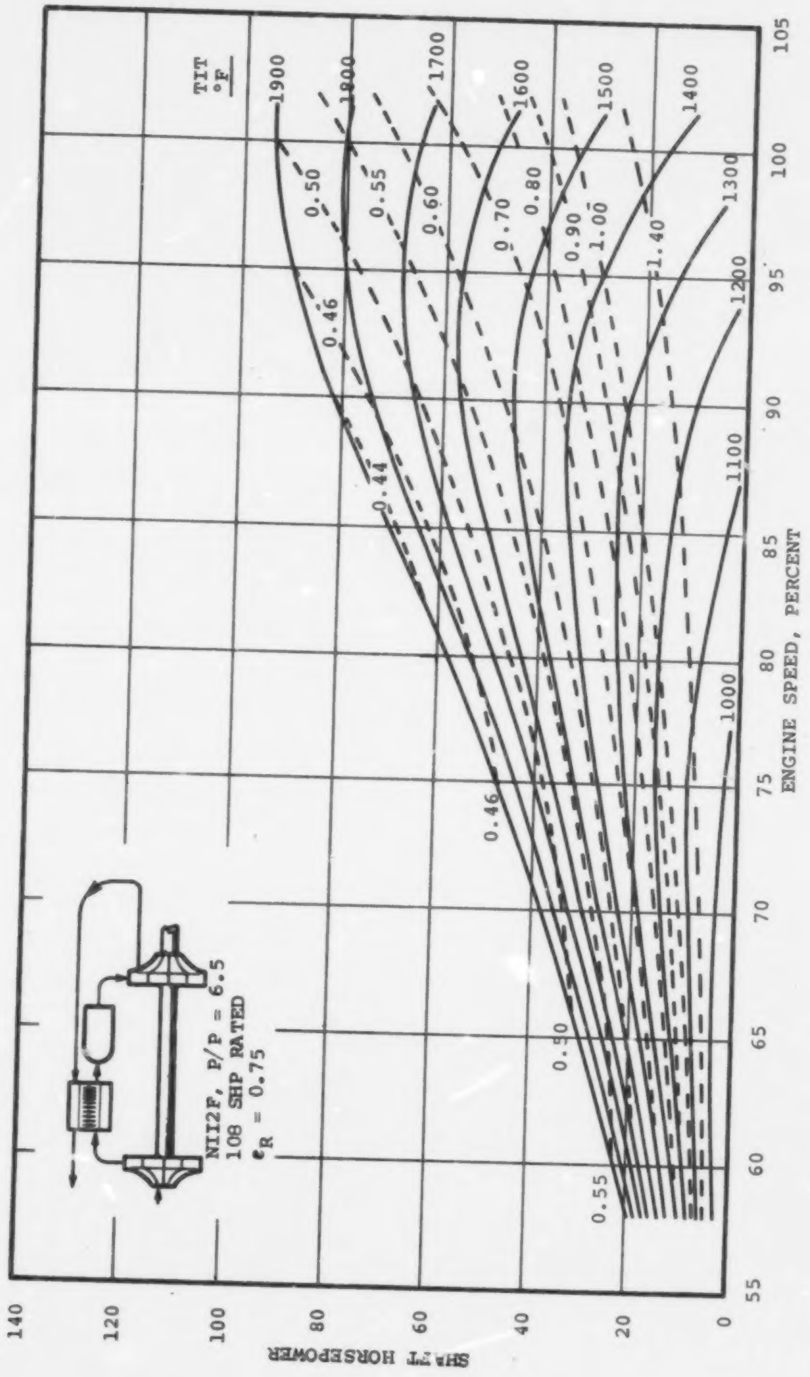
SINGLE-SHAFT CYCLE, 85°F DAY

FIGURE 4-10



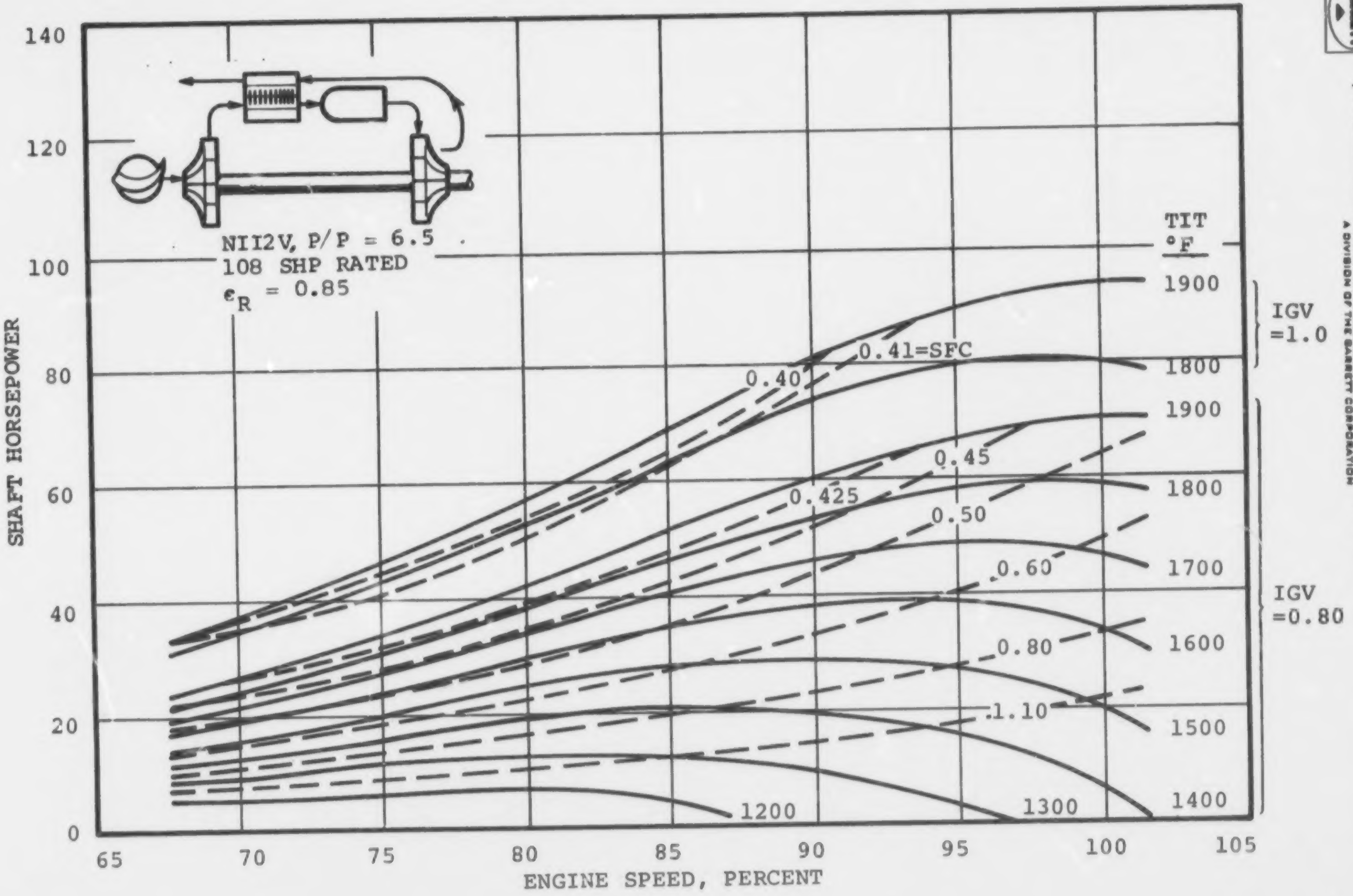
RECUPERATED SINGLE-SHAFT CYCLE, 85°F DAY

FIGURE 4-11



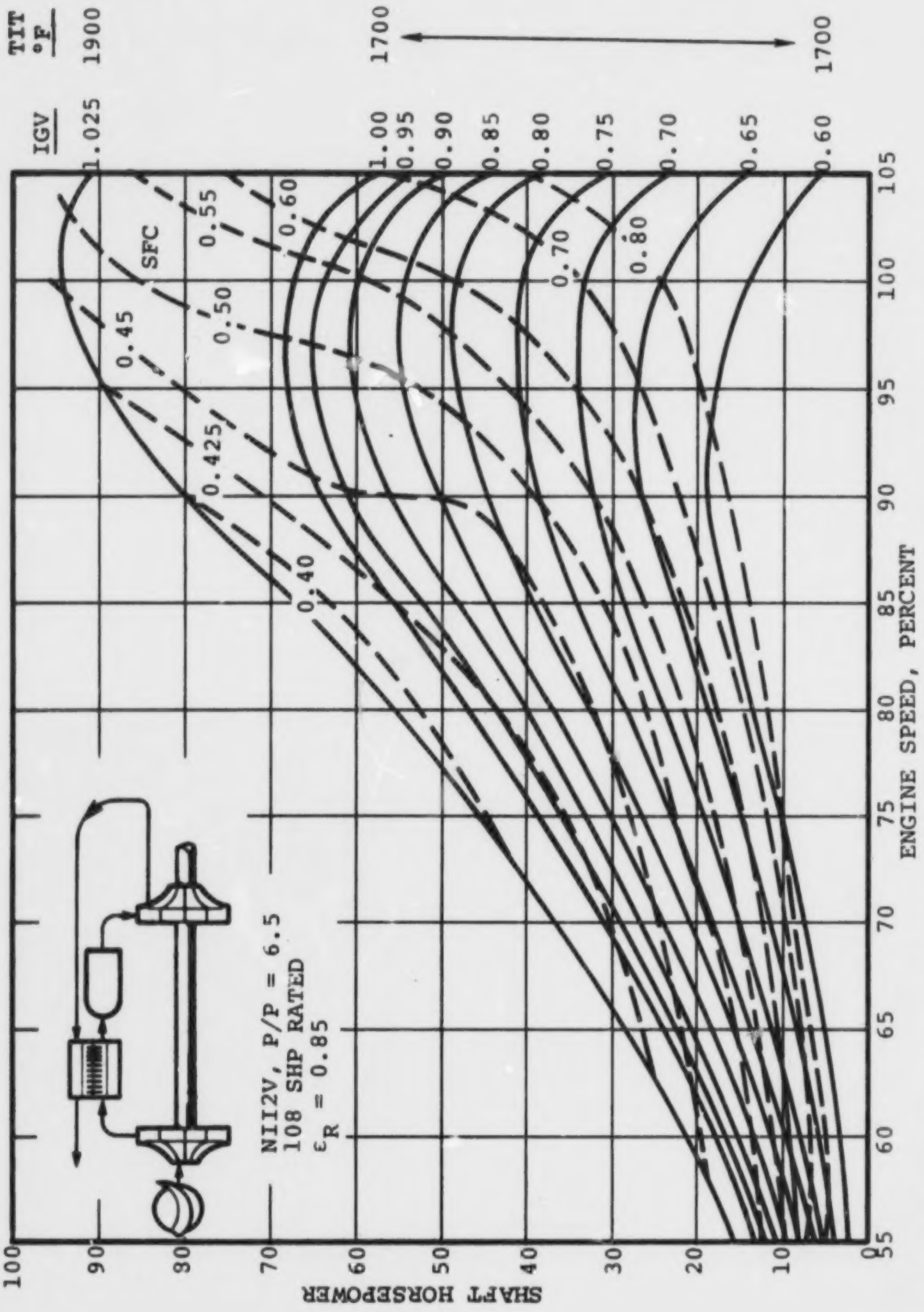
RECUPERATED SINGLE-SHAFT CYCLE, 85°F

FIGURE 4-12



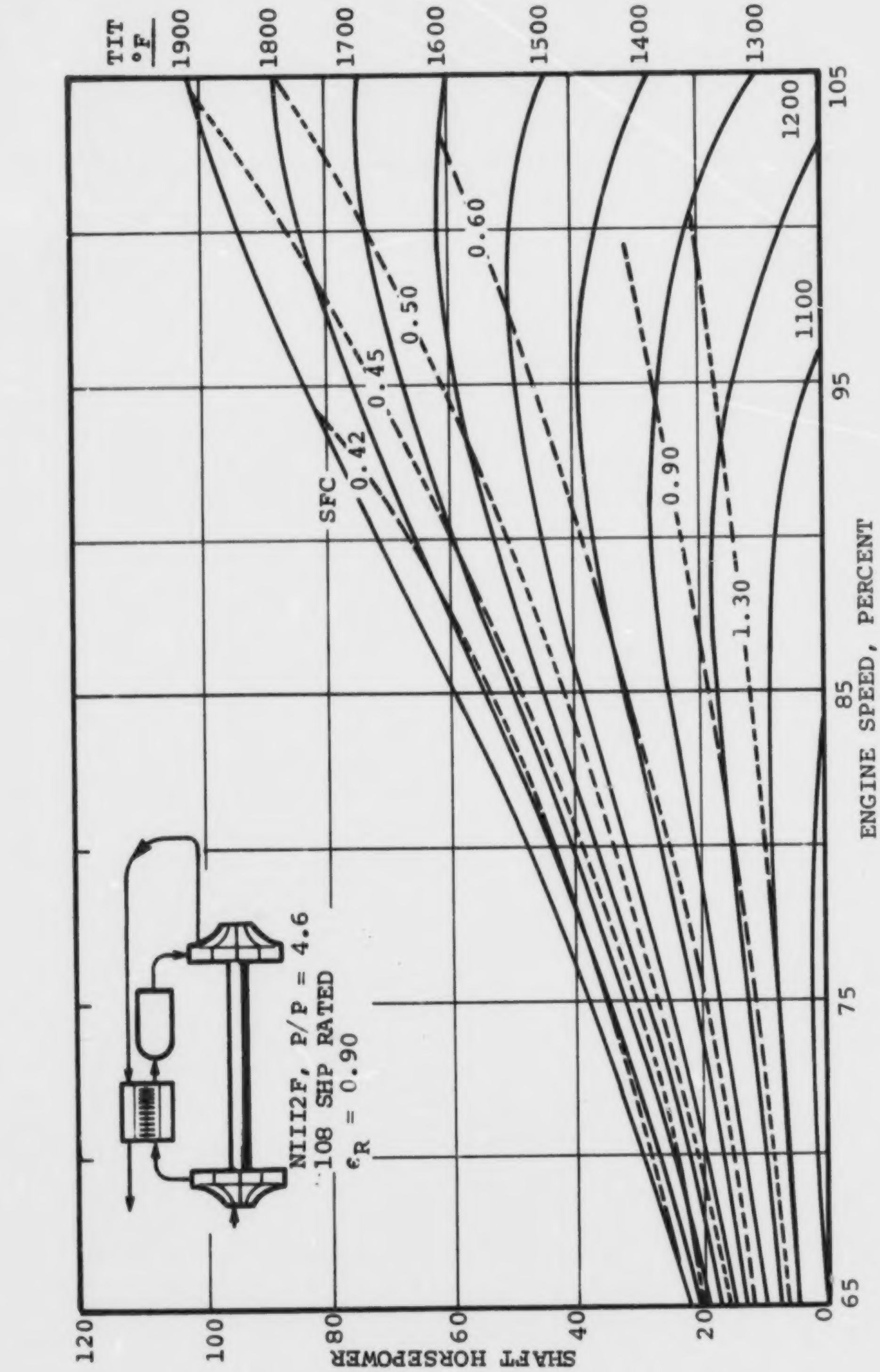
RECUPERATED SINGLE-SHAFT CYCLE WITH VARIABLE INLET GUIDE VANES, 85°F DAY

FIGURE 4-13



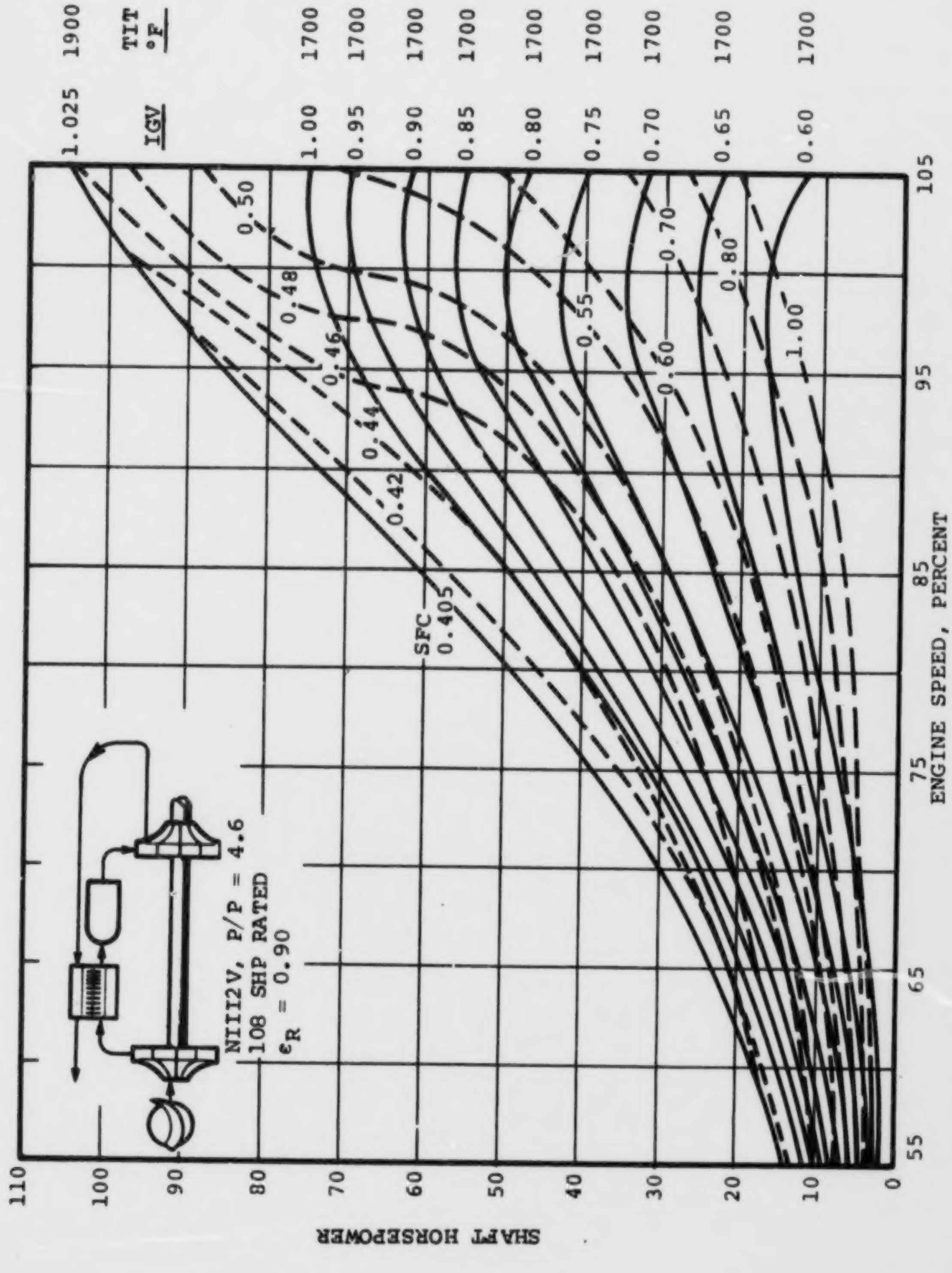
RECUPERATED SINGLE-SHAFT CYCLE WITH VARIABLE INLET GUIDE VANES, 85°F DAY

FIGURE 4-14



REGENERATED SINGLE-SHAFT CYCLE, 85°F DAY

FIGURE 4-15



REGENERATED SINGLE-SHAFT CYCLE WITH VARIABLE INLET GUIDE VANES, 85°F DAY

FIGURE 4-16

(a) Recuperator (fixed boundary)

(b) Regenerator (rotary)

(c) Variable area power turbine nozzles

(d) Recuperator bypass

(e) Power transfer schemes

(f) Variable inlet guide vanes

Based on the results obtained from the mission analysis investigation (Sections 6 and 7) of performance exhibited by the various engine configurations and such factors as size, weight, complexity, and relative cost, a selection of three candidate cycles was made. A detailed off-design performance analysis was conducted on these candidate engines and is discussed in Section 5 of this report.

SECTION 5

DETAILED OFF-DESIGN PERFORMANCE OF THE THREE SELECTED CANDIDATE CYCLES

The three candidate gas turbine cycles chosen for detailed off-design analysis (Section 7) are described as follows:

(a) NII2V - Recuperated single-shaft cycle with variable inlet guide vanes:

Compressor	Single-stage centrifugal
Pressure ratio at design power	6.4
Variable inlet vanes	IGV angle setting = 1.025 to 0.60 $= \dfrac{\frac{W\sqrt{\theta}}{\delta}\ \text{(with swirl)}}{\frac{W\sqrt{\theta}}{\delta}\ \text{(zero swirl)}}$
Turbine	Single-stage radial
Recuperator	Fixed boundary, counterflow (ceramic and metallic)
Sea-level, standard-day rating	
Design power	108 shp, scaled to 135

Transient power increased by water injection	$\frac{(\mathrm{shp})_{\mathrm{wet}}}{(\mathrm{shp})_{\mathrm{dry}}} = 1.30$ maximum
Recuperator effectiveness	$\epsilon_R = 0.85$

Inlet guide vanes were added to the cycle to improve part-load specific fuel consumption, especially at idle conditions.

(b) NIII2V - Regenerated single-shaft cycle with variable inlet guide vanes:

Compressor	Single-stage centrifugal
Pressure ratio design power	4.6
Variable inlet guide vanes	IGV angle setting $= 1.025$ to 0.60 $= \dfrac{\frac{W\sqrt{\theta}}{\delta}\ (\text{with swirl})}{\frac{W\sqrt{\theta}}{\delta}\ (\text{zero swirl})}$
Turbine	Single-stage radial
Regenerator	Ceramic rotary type
Sea-level, standard day rating	
Design power	108 shp

Transient power increased by water injection	$\frac{(\text{shp})_{\text{wet}}}{(\text{shp})_{\text{dry}}} = 1.30$ maximum
Without power autmentation	155 shp
Regenerator	$\varepsilon_R = 0.90$

(c) AIII2V - Regenerated free-turbine cycle with variable power turbine nozzles and inlet guide vanes:

Compressor	Single-stage centrifugal
Pressure ratio at design power	4.6
Variable inlet guide vanes	IGV angle setting $= 1.0$ to 0.50 $$= \frac{\frac{W\sqrt{\theta}}{\delta}\ \text{(with swirl)}}{\frac{W\sqrt{\theta}}{\delta}\ \text{(zero swirl)}}$$
Gasifier turbine	Single-stage radial
Power turbine	Single-stage axial
Variable power turbine nozzle	A_N/A_n design $= 1.0$ to 0.70
Regenerator	Ceramic rotary type

Sea-level, standard-day rating

Design power	175 shp
Corresponding minimum gas generator idle speed	73 percent of design speed
With water injection for power augmentation	135 shp

The optimum pressure ratio of an automobile gas turbine is not necessarily the pressure ratio which yields the minimum specific fuel consumption (sfc) at all load conditions. Cycle complexity, ruggedness, reliability, and component costs must also be considered. The following characteristics were qualitatively evaluated in the selection of cycle pressure ratio:

(a) Heat Exchanger - with increased pressure ratio:

  (1) Optimum heat recovery effectiveness decreases

  (2) Size and costs decrease because of the higher specific density and specific power

  (3) Seal leakage increases

(b) Compressor - with increased pressure ratio:

  (1) Attainable efficiency decreases (Section 2.1.1)

  (2) More critical impeller and diffuser matching because of higher Mach number levels

(3) Speed and stresses increase

(4) Material costs increase

(5) Changing aluminum to stainless steel increases inertia

(c) Turbine - with increased pressure ratio:

(1) Turbine rotor stresses and shaft speed increase

(2) Component size decreases

(3) Optimum turbine inlet temperature increases (Section 3)

(4) Material costs increase because of higher temperature requirement

(d) Bearings and Seals - with increased pressure ratio:

(1) Bearing reliability decreases because of higher speeds

(2) Seal requirements and cost increase

The parametric cycle studies (Section 3) and experience were used in evaluating the above effects to obtain the proper range of cycle pressure ratios for a given heat recovery effectiveness. Because of the number of factors that must be considered, there exists no simple means of determining the optimum pressure ratio. As a result, several values in the range determined by the above criteria were evaluated for a given effectiveness value. Off-design performance was computed for each of the cycles and used in the mission analysis program to compute fuel economy values. Then fuel economy, heat exchanger size, weight, and estimated cost were evaluated (as well as shaft speeds, component stresses, and material requirements) for selecting a pressure ratio for each of the engines considered in the study.

5.1 CYCLE DESIGN POINT

Design point calculations for the three selected cycles are shown on Tables 5-1, 5-2, and 5-3. Note that the maximum temperature for the single-shaft cycles is 1900°F, whereas the free-turbine has a maximum cycle temperature of 1850°F. If all three cycles had the same turbine inlet total gas temperature, the average for the gas generator turbine blades would be higher for the free-turbine cycle because of its lower pressure ratio. Consequently, the maximum temperature of the free-turbine cycle was selected as 1850°F.

The fuel lower heating value was degraded from 18,500 to 18,300 Btu/lb to account for radiation heat loss to the cycle.

5.2 COMPONENT MAPS

5.2.1 Combustor Maps

Combustor maps used in the off-design calculations for the three engine configurations are shown in Figures 5-1, 5-2, and 5-3. The combustors were designed for a part-load operating condition rather than for maximum power, to ensure that the required high-efficiency levels will be achieved at near-idle conditions.

5.2.2 Compressor Maps

Compressor performance maps with engine operating lines are shown in Figures 5-4, 5-5, and 5-6. These maps were scaled from data of existing components, having pressure ratio and Mach number levels comparable to the candidate cycles. Note that the maximum power match-point was selected so that maximum compressor efficiency was achieved at low, part-load conditions. The compressor utilizes backward-curved impeller blades that shift the peak efficiency envelope away from the stall-limit line, thus providing ample stall-margin with respect to the engine-operating line.

TABLE 5-1

RECUPERATED SINGLE-SHAFT CYCLE
(ENGINE NII2V)

LHV	H/C	ALTITUDE	FPS
18300.0	0.16786	0.0	0.000

COMPRESSOR			
DRIVING TURBINE	HP	ENERGY CORR. COOLING AIR	WATER
1	162.69	0.000	0.000

TURBINE					
				BEFORE MIXING	
SHAFT HP	ACC HP	HP REQUIRED	MECHANICAL EFFICIENCY	ENTHALPY	TEMP
108.0	6.0	279.0	0.980	425.10	1687.5

BURNER	WATER	FUEL FLOW
	0.000	46.89

	CR FLOW	PRESSURE	TEMP	DELTA	THETA	R	ENTHALPY	GAMMA	F/A	W/A	EFF	P/P	DEL P	LEAKAGE	COOLING
AMBIENT	1.043	14.696	518.7	1.000	1.000	53.349	124.00	1.401	--	0.0000	--	--	--	--	--
INLET	1.059	14.476	518.7	0.985	1.000	53.349	124.00	1.401	--	0.0000	1.000	1.000	0.015	0.000	--
COMPRESSR	0.221	92.861	972.7	6.319	1.875	53.349	234.28	1.382	--	0.000	0.790	6.415	0.000	0.006	0.015
DIFFUSER	0.223	91.932	972.7	6.256	1.875	53.349	234.28	1.382	0.0000	0.0000	--	--	0.010	--	--
REGNR CLD	0.291	89.818	1571.4	6.112	3.030	53.349	388.11	1.347	--	0.0000	0.850	1.000	0.023	0.000	--
BURNER	0.376	86.225	2359.7	5.867	4.549	53.371	615.85	1.308	0.0128	0.0000	0.999	1.000	0.040	0.000	--
TURBINE	1.638	16.932	1677.5	1.152	3.234	53.371	422.26	1.333	0.0126	0.000	[illegible]75	5.092	0.000	0.000	0.015
DIFFUSER	1.743	15.916	1677.5	1.083	3.234	53.371	422.26	1.333	0.0126	0.0000	--	--	0.060	--	--
REGRN HOT	1.522	14.850	1114.5	1.010	2.149	53.371	272.63	1.365	0.0126	0.0000	--	1.000	0.067	0.000	--
DIFFUSER	1.538	14.700	1114.5	1.000	2.149	53.371	272.63	1.365	0.0126	0.0000	--	--	0.010	--	--

FUEL FLOW	TOTAL	CORRECTED	SPC
	46.9	47.6	0.434

POWER	NET	CORRECTED	SPECIFIC
	108.0	109.6	103.57

TABLE 5-2

REGENERATED SINGLE-SHAFT CYCLE
(ENGINE NIII2V)

LHV	B/C	ALTITUDE	PPS
18300.0	0.16786	0.0	0.000

COMPRESSOR			
DRIVING TURBINE	HP	ENERGY CORR. COOLING AIR	WATER
1	142.76	0.000	0.000

TURBINE					
				BEFORE MIXING	
SHAFT HP	ACC HP	HP REQUIRED	MECHANICAL EFFICIENCY	ENTHALPY	TEMP
108.0	6.0	259.1	0.980	453.87	1794.8

BURNER	WATER	FUEL FLOW
	0.000	44.29

	CR FLOW	PRESSURE	TEMP	DELTA	THETA	R	ENTHALPY	GAMMA	F/A	W/A	EFF	P/P	DEL P	LEAKAGE	COOLING
AMBIENT	1.218	14.696	518.7	1.000	1.000	53.349	124.00	1.401	--	0.0000	--	--	--	--	--
INLET	1.237	14.476	518.7	0.985	1.000	53.349	124.00	1.401	--	0.0000	1.000	1.000	0.015	0.000	--
COMPRESSR	0.339	66.689	861.2	4.538	1.660	53.349	206.81	1.389	--	0.0000	0.822	4.607	0.000	0.006	0.015
DIFFUSER	0.342	66.022	861.2	4.493	1.660	53.349	206.81	1.389	0.0000	0.0000	--	--	0.010	--	--
REGNR CLD	0.464	64.504	1689.6	4.389	3.257	53.349	419.79	1.341	--	0.0000	0.900	1.000	0.023	0.027	--
BURNER	0.577	61.923	2359.7	4.214	4.549	53.368	614.40	1.310	0.0109	0.0000	0.999	1.000	0.040	0.000	--
TURBINE	1.855	16.924	1781.1	1.152	3.434	53.367	449.98	1.329	0.0107	0.0000	0.890	3.659	0.000	0.000	0.016
DIFFUSER	1.984	15.908	1781.1	1.082	3.434	53.367	445.98	1.329	0.0107	0.0000	--	--		--	--
REGRN HOT	1.674	14.843	990.9	1.010	1.910	53.366	240.73	1.374	0.0102	0.0000	--	1.000	0.067	0.027	--
DIFFUSER	1.691	14.694	990.9	1.000	1.910	53.366	240.73	1.374	0.0102	0.0000	--	--	0.010	--	--

FUEL FLOW	TOTAL	CORRECTED	SFC
	44.3	45.0	0.410

POWER	NET	CORRECTED	SPECIFIC
	108.0	109 6	88.64

TABLE 5-3

REGENERATED FREE-TURBINE CYCLE
(ENGINE AIII2V)

LHV	H/C	ALTITUDE	FPS
18300.0	0.16786	0.0	0.000

COMPRESSOR

DRIVING TURBINE	HP	ENERGY CORR. COOLING AIR	WATER
1	212.00	0.000	0.000

TURBINES
(2 IS THE POWER TURBINE)

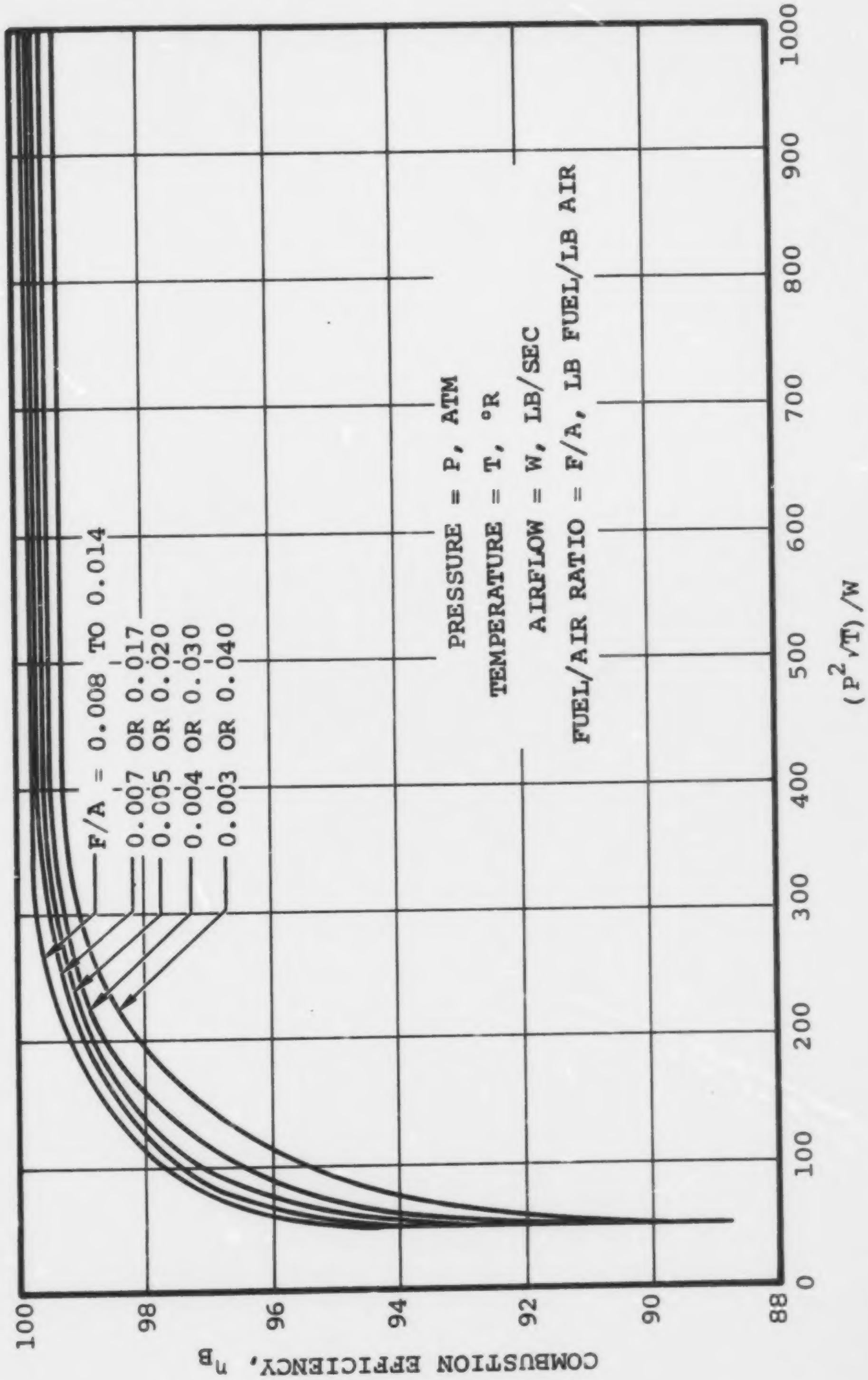
SHAFT HP	ACC HP	HP REQUIRED	MECHANICAL EFFICIENCY	BEFORE MIXING ENTHALPY	BEFORE MIXING TEMP
0.0	5.0	217.1	0.980	509.07	1992.8
150.0	1.5	154.6	0.980	440.85	1748.8

BURNER	WATER	FUEL FLOW
	0.000	63.41

	CR FLOW	PRESSURE	TEMP	DELTA	THETA	R	ENTHALPY	GAMMA	F/A	W/A	EFF	P/P	DEL P	LEAKAGE	COOLING
AMBIENT	1.805	14.596	518.7	1.000	1.000	53.349	124.00	1.401	--	0.0000	--	--	--	--	--
INLET	1.833	14.476	518.7	0.985	1.000	53.349	124.00	1.401	--	0.0000	1.000	1.000	0.015	0.000	--
COMPRESSR	0.509	65.980	862.0	4.490	1.662	53.349	207.01	1.389	--	0.0000	0.813	4.558	0.000	0.003	0.015
DIFFUSER	0.514	65.320	862.0	4.445	1.662	53.349	207.01	1.389	--	0.0000	--	--	0.010	--	--
REGNR CLD	0.691	63.817	1660.6	4.343	3.201	53.349	411.97	1.342	0.0000	0.0000	0.900	1.000	0.023	0.027	--
BURNER	0.858	61.265	2309.7	4.169	4.453	53.367	599.62	1.311	--	0.0000	0.999	1.000	0.040	0.000	--
TURBINE	1.633	30.239	1976.3	2.058	3.810	53.367	504.32	1.322	0.0105	0.0000	0.880	2.026	0.000	0.000	0.016
DIFFUSER	1.667	29.634	1976.3	2.016	3.810	53.367	504.32	1.322	0.0103	0.0000	--	--	0.020	--	--
TURBINE	2.773	16.756	1748.8	1.140	3.371	53.367	440.85	1.331	0.0103	0.0000	0.880	1.769	0.000	0.000	0.000
DIFFUSER	2.950	15.750	1748.8	1.072	3.371	53.367	440.85	1.331	0.0103	0.0000	--	--	0.060	--	--
REGRN HOT	2.506	14.695	985.9	1.0[illegible]0	1.901	53.366	239.41	1.375	0.0098	0.0000	--	1.000	0.067	0.027	--

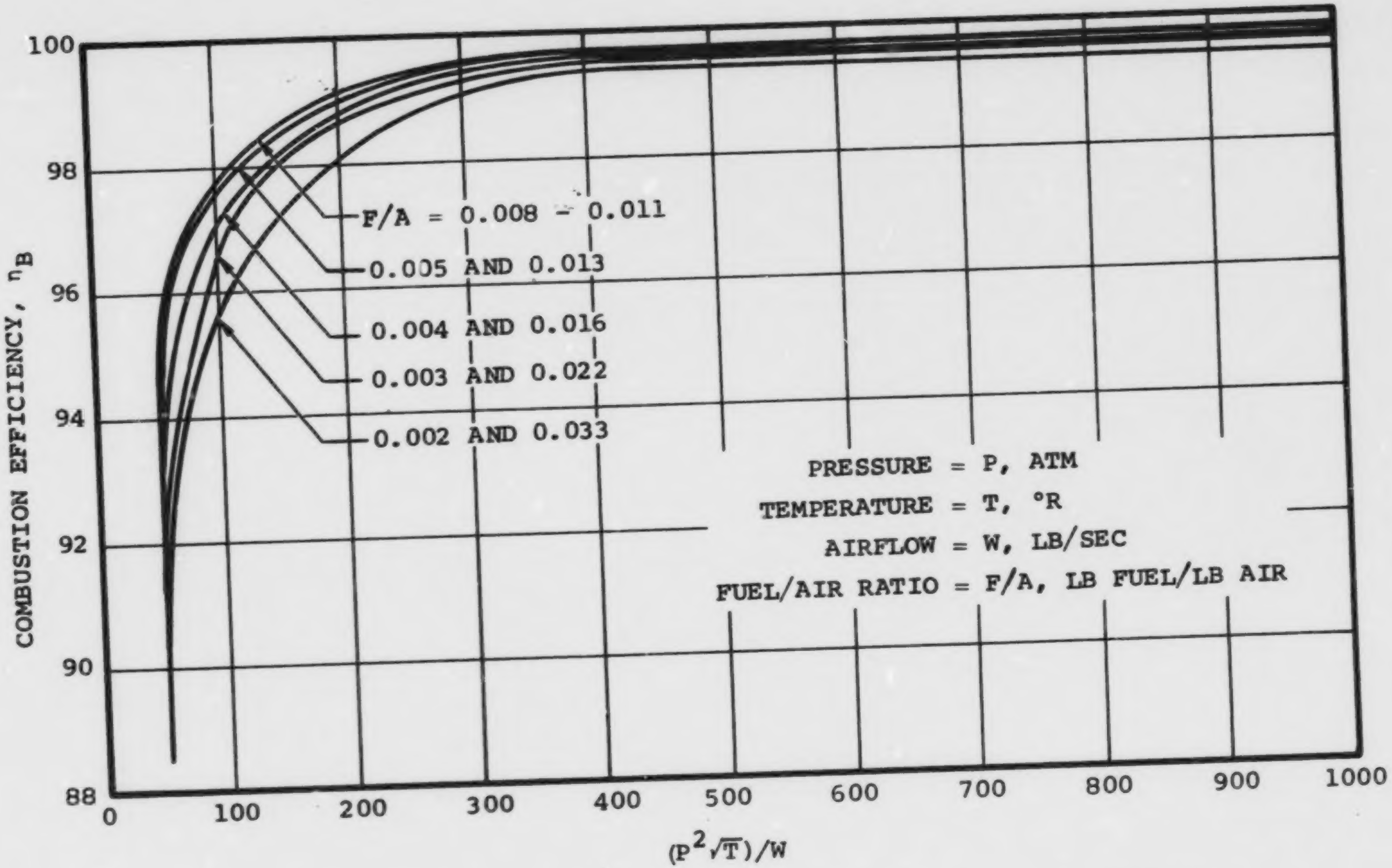
FUEL FLOW	TOTAL	CORRECTED	SFC
	63.4	64.4	0.423

POWER	NET	CORRECTED	SPECIFIC
	150.0	152.3	83.10



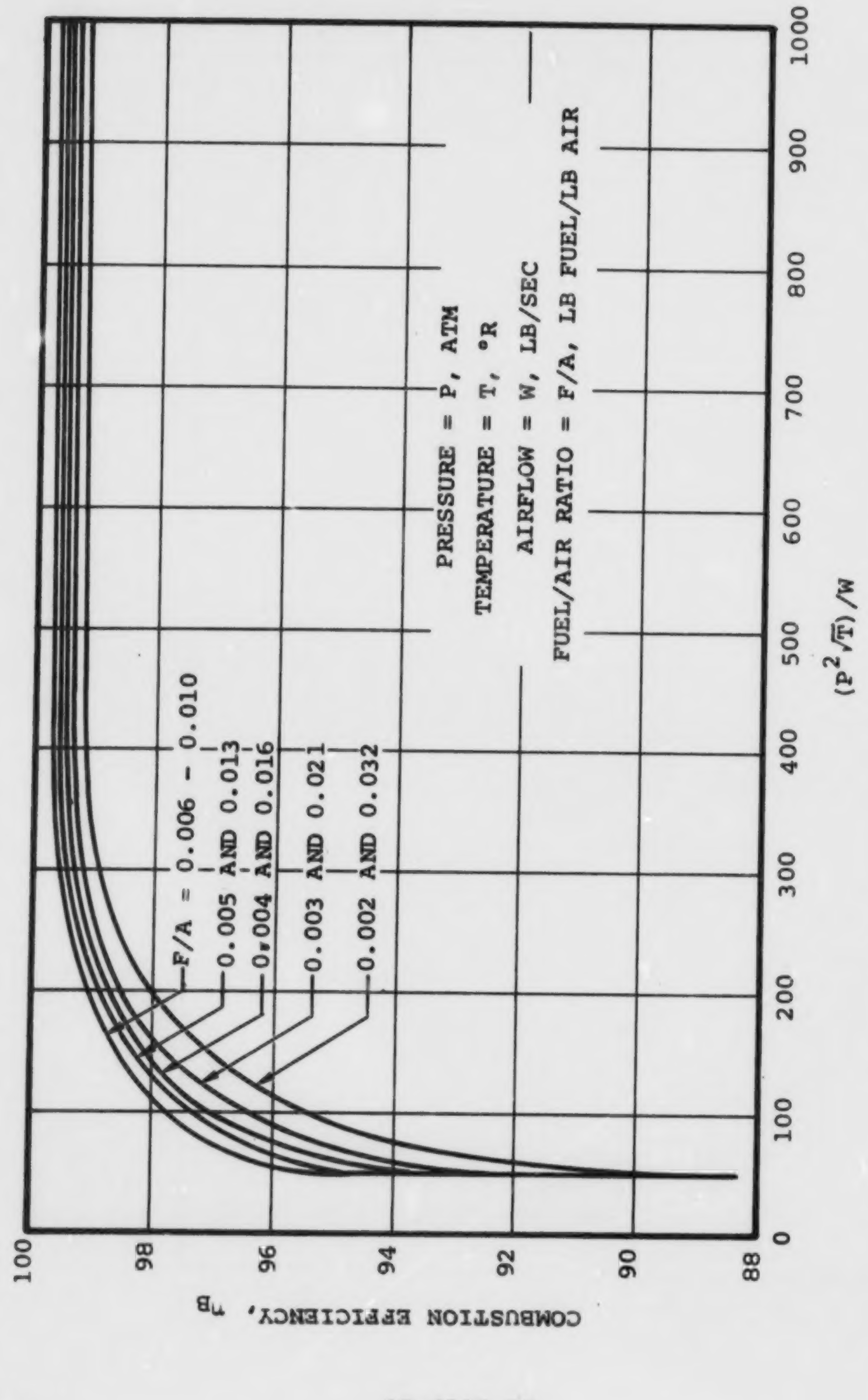
SINGLE-SHAFT RECUPERATED ENGINE (NII2V) COMBUSTOR EFFICIENCY

FIGURE 5-1



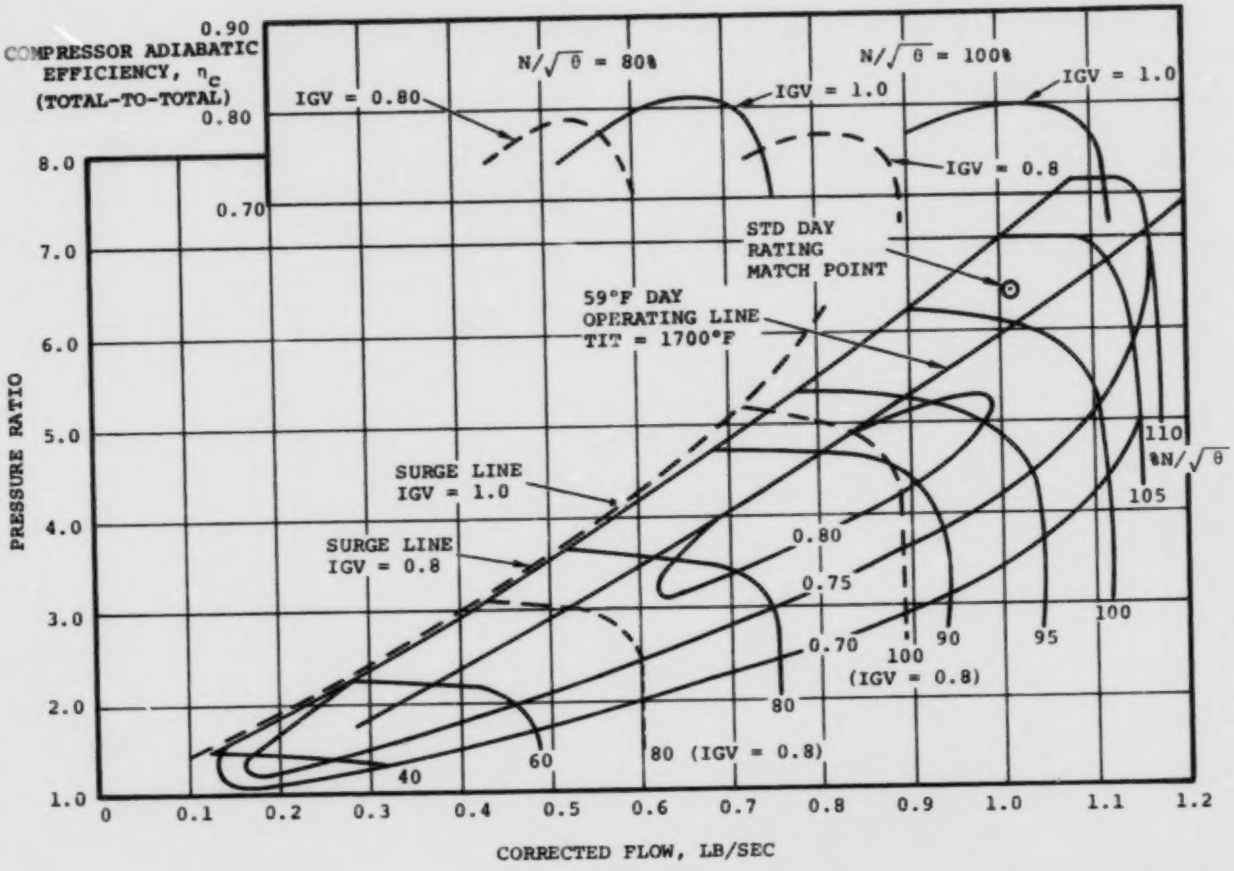
SINGLE-SHAFT REGENERATED ENGINE (NIII2V) COMBUSTOR EFFICIENCY

FIGURE 5-2



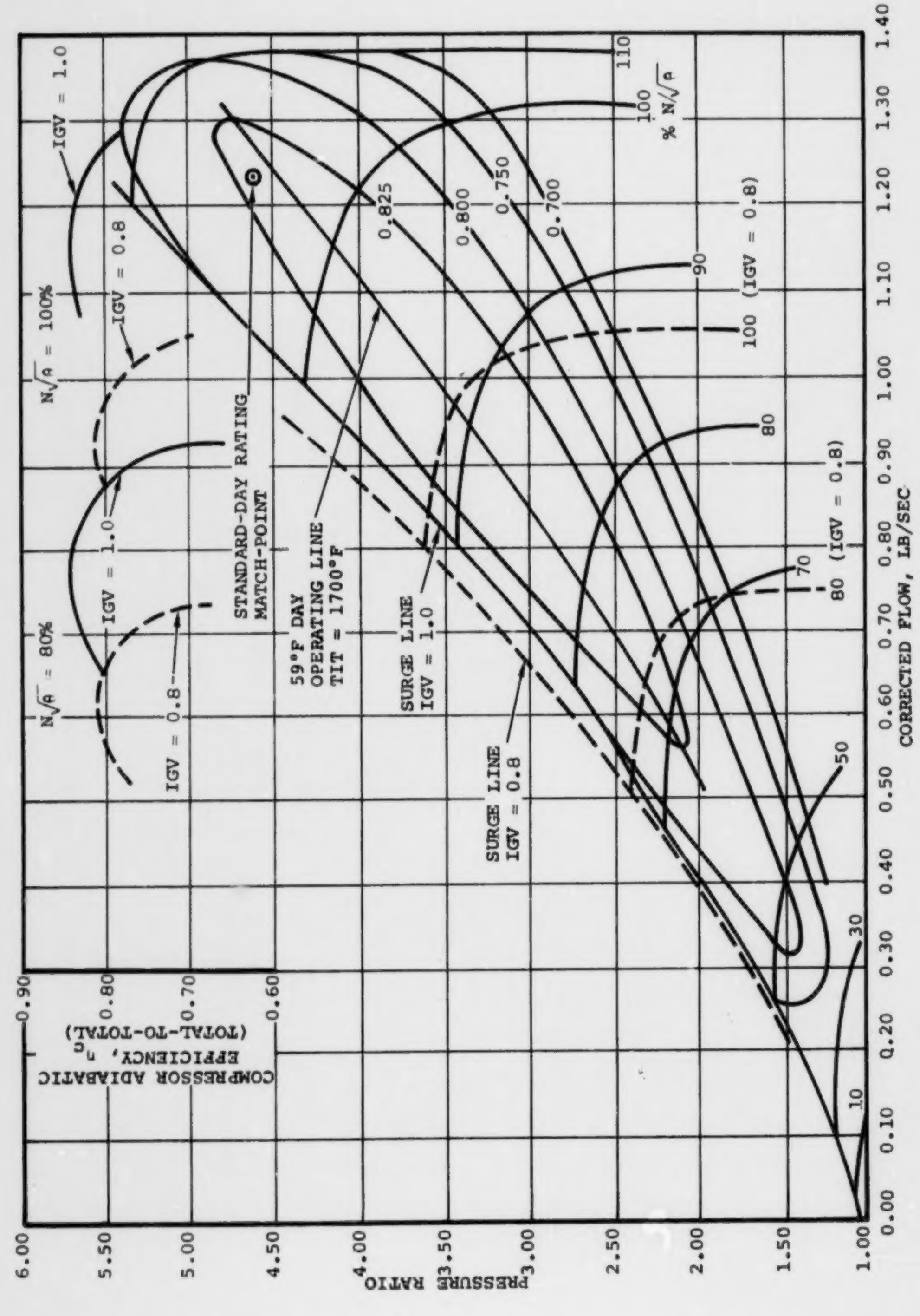
FREE-TURBINE REGENERATED ENGINE (AIII2V) COMBUSTOR EFFICIENCY

FIGURE 5-3



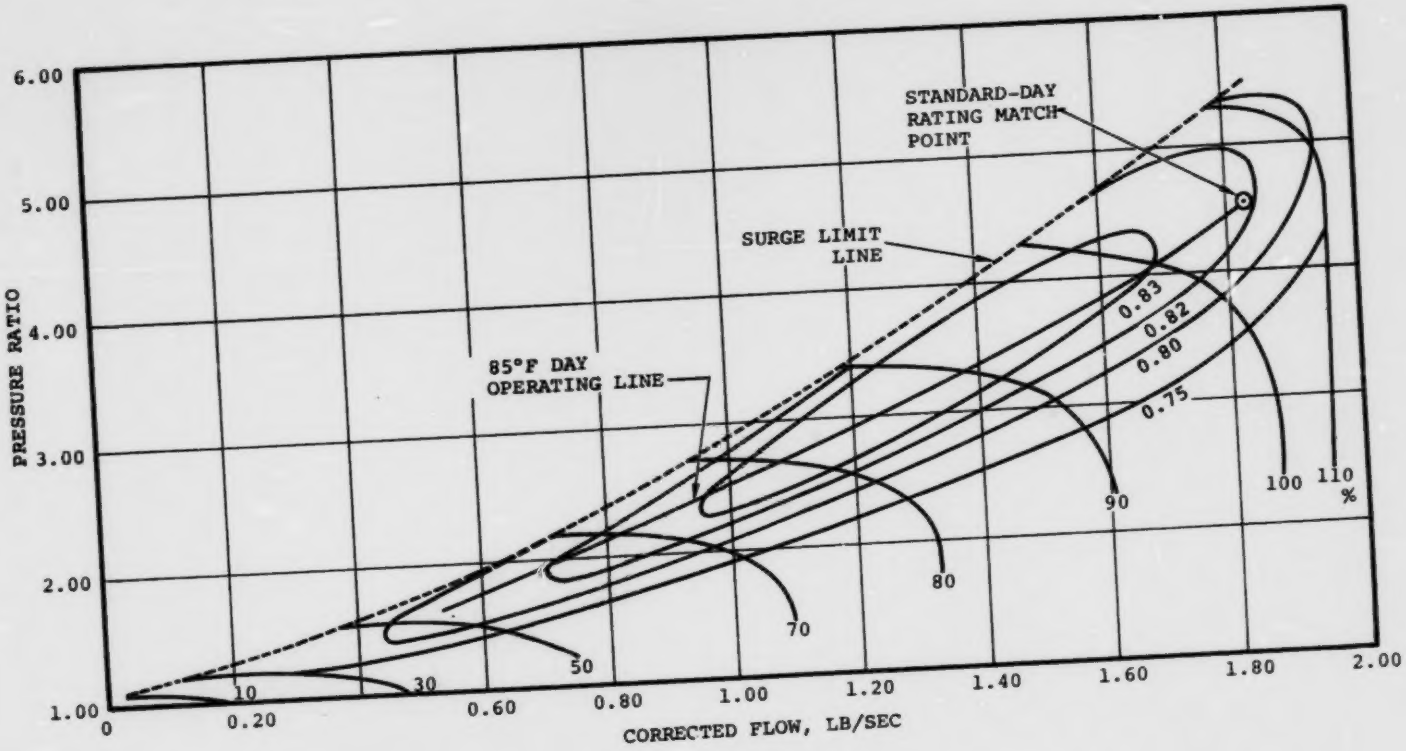
SINGLE-SHAFT RECUPERATED ENGINE (NII2V) COMPRESSOR PERFORMANCE

FIGURE 5-4



SINGLE-SHAFT REGENERATED ENGINE (NIII2V) COMPRESSOR PERFORMANCE

FIGURE 5-5



FREE-TURBINE REGENERATED ENGINE (AIII2V) COMPRESSOR PERFORMANCE

FIGURE 5-6

Since an infinite number of maps could be shown for compressors with variable inlet guide vanes, only two settings are superimposed on the single-shaft compressor maps as an illustration. The vane setting of IGV = 1.0 refers to zero inlet swirl; that of IGV = 0.80 refers to a setting which induces swirl so that the ratio of flow-with-swirl to flow-with-zero-swirl is determined:

$$\frac{\frac{W\sqrt{\theta}}{\delta}\ \text{(with swirl)}}{\frac{W\sqrt{\theta}}{\delta}\ \text{(zero swirl)}} = 0.8$$

5.2.3 Turbine Maps

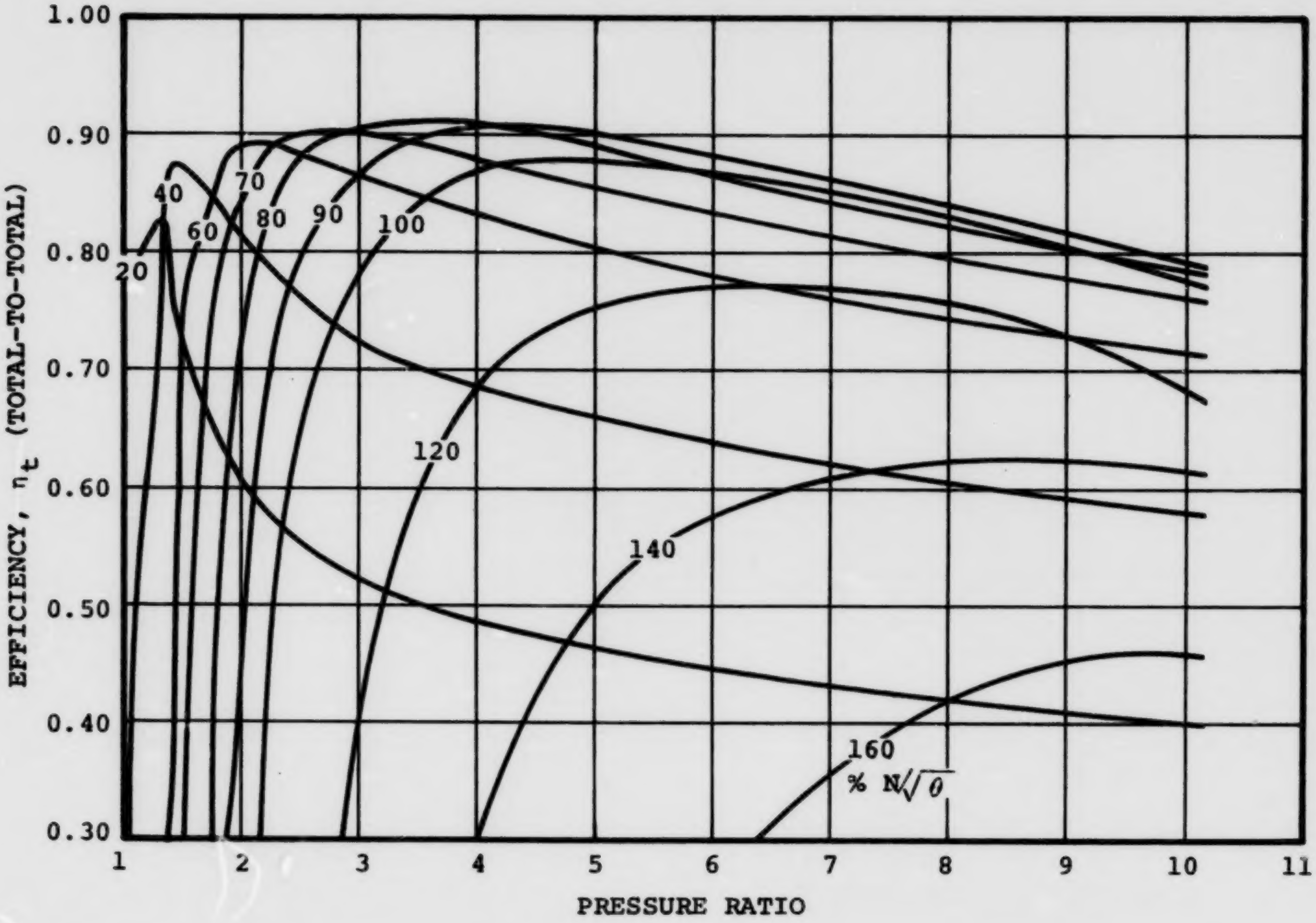
Radial turbine maps for the two single-shaft cycles and the radial gas generator turbine of the free-turbine cycle are shown in Figures 5-7 through 5-12. Figures 5-13 and 5-14 show typical maps for one setting of the variable area power turbine nozzle; however, 10 different maps, having various nozzle area settings (0.5 to 1.0), were used in the off-design computer program.

Efficiency and pressure-flow characteristics as a function of corrected speed are scaled from experimental data of existing turbines.

5.2.4 Heat-Exchanger Maps

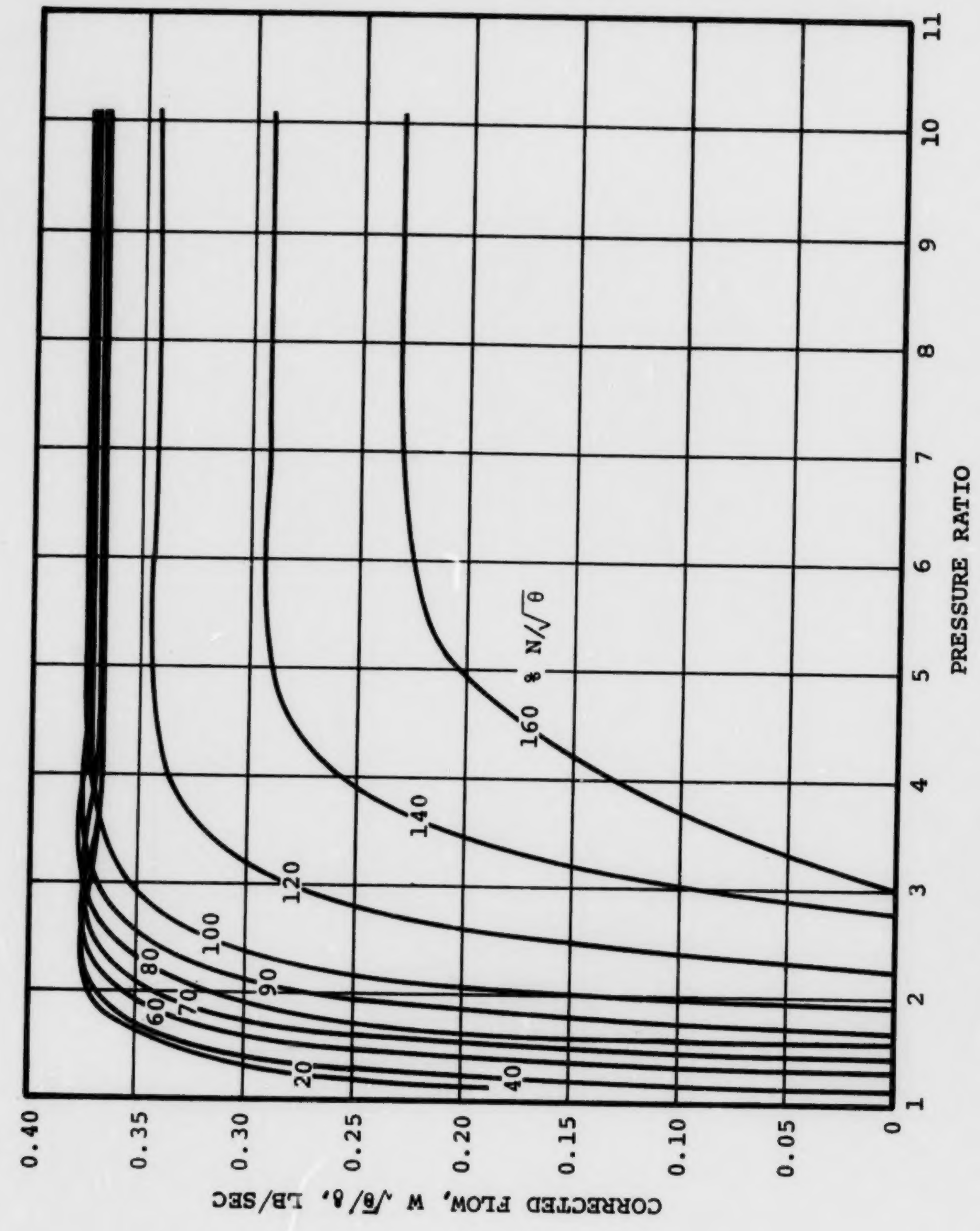
Estimated off-design performance of the ceramic rotary regenerator is shown in Figures 5-15 and 5-16. In addition to the core fractional pressure loss (Figure 5-16), a total of 3 percent duct pressure loss was used. This was varied as a function of the square of the corrected flow into the ducts.

The regenerator clearance and seals will be sized for the static and carryover design leakage goals shown in Figure 5-17.[8,9] The leakage at off-design conditions was varied by approximate linearity with pressure.



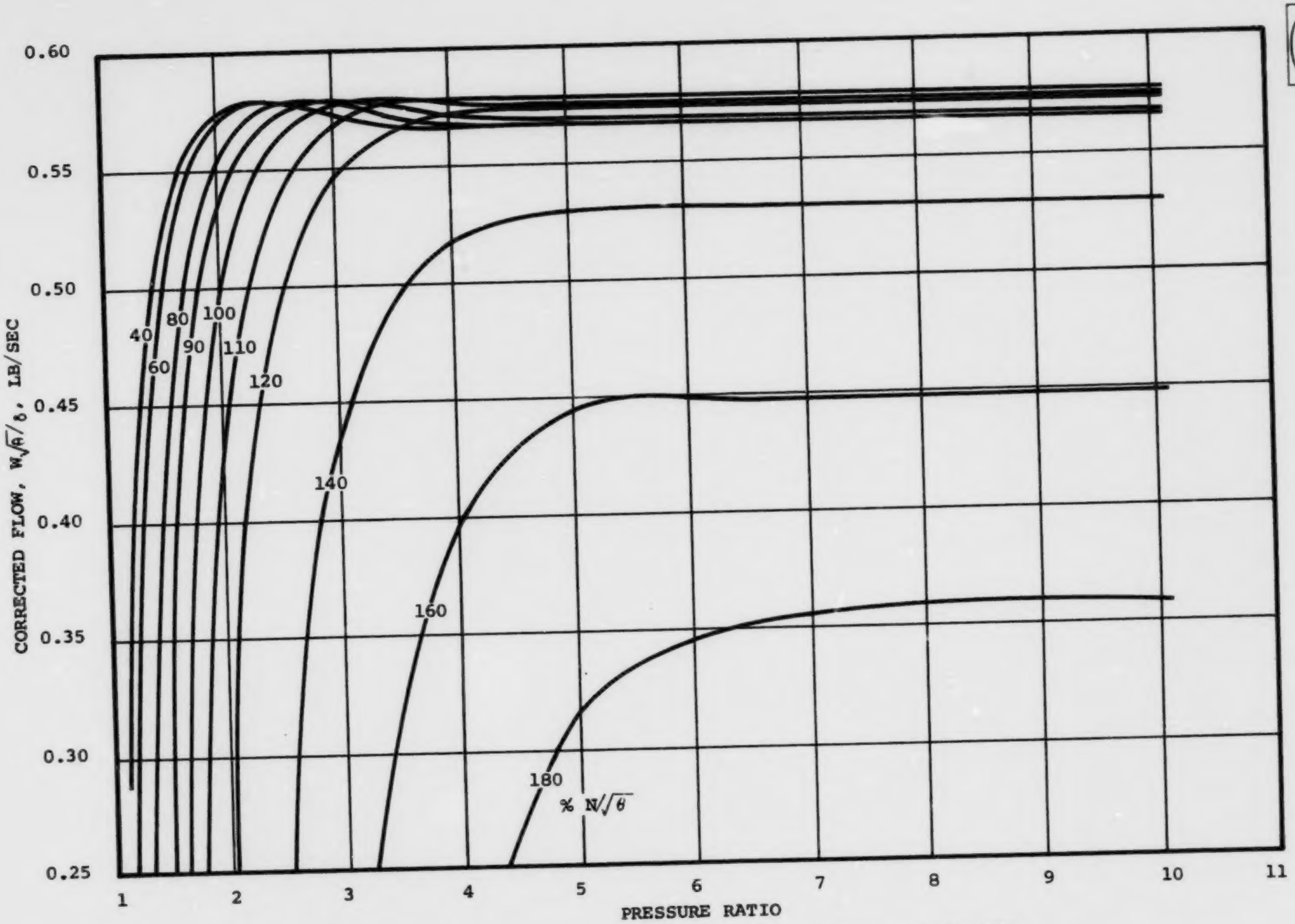
SINGLE-SHAFT RECUPERATED ENGINE (NII2V) TURBINE PERFORMANCE

FIGURE 5-7



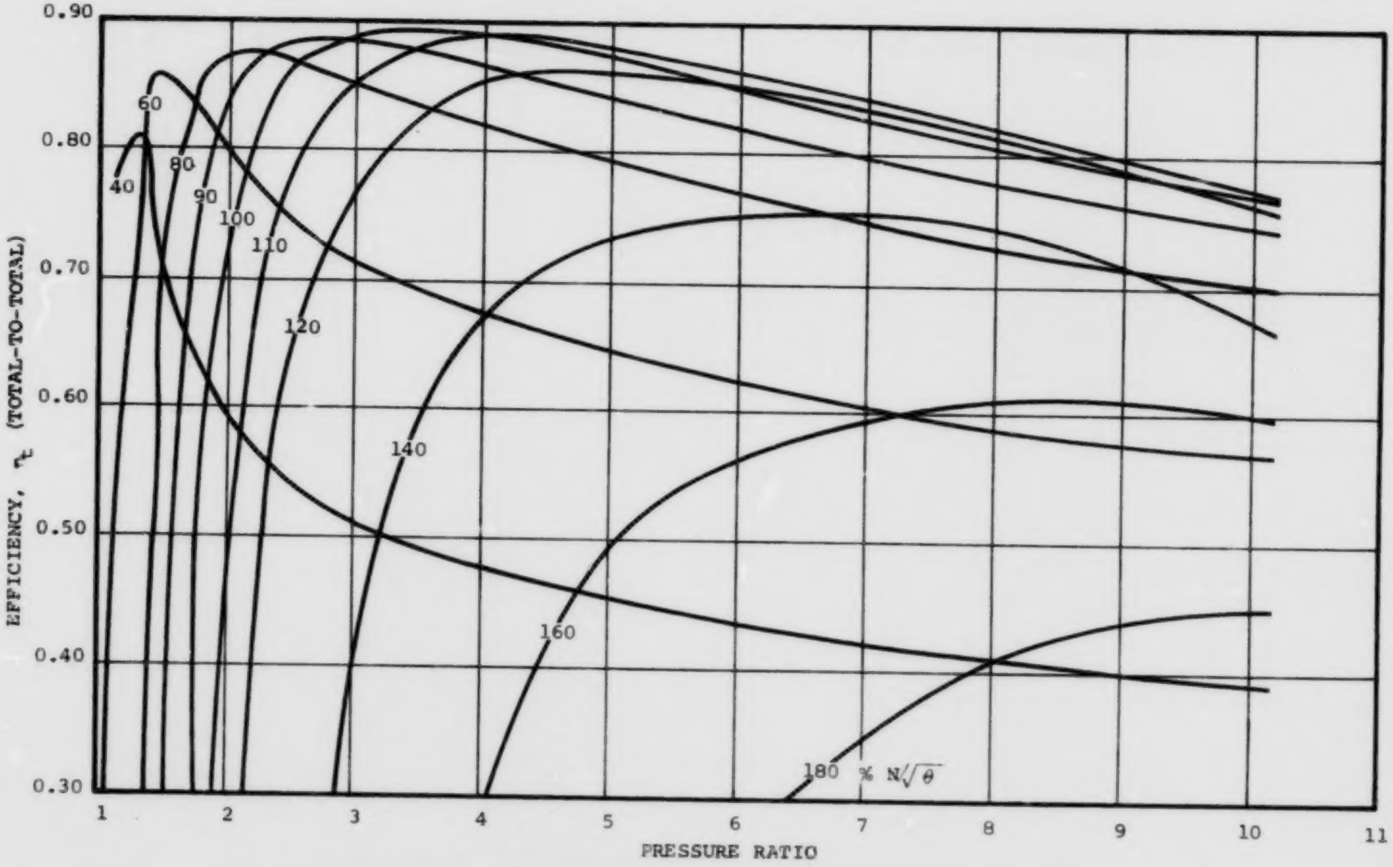
SINGLE-SHAFT RECUPERATED ENGINE (NII2V) TURBINE PERFORMANCE

FIGURE 5-8



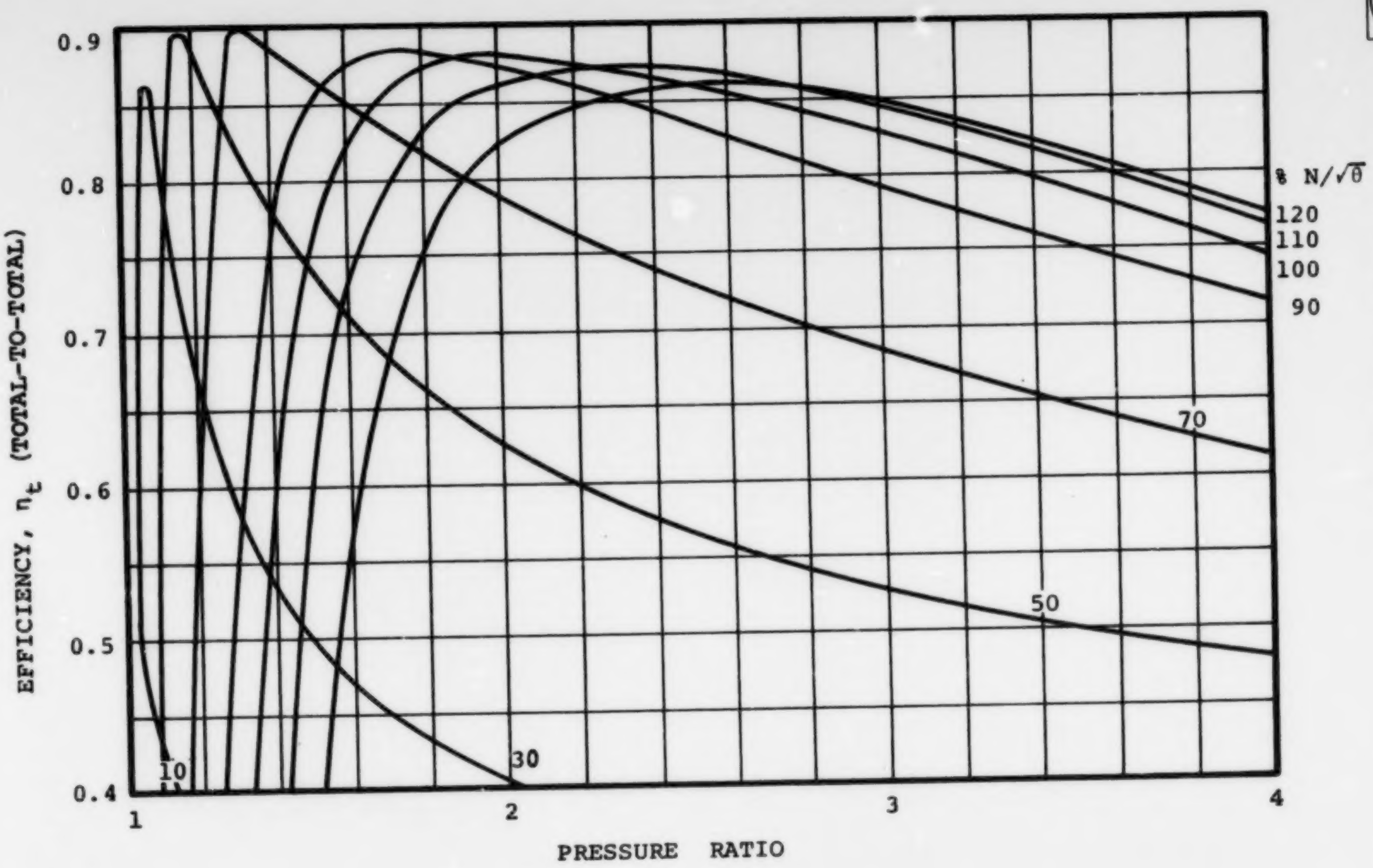
SINGLE-SHAFT REGENERATED ENGINE (NIII2V) TURBINE PERFORMANCE

FIGURE 5-9



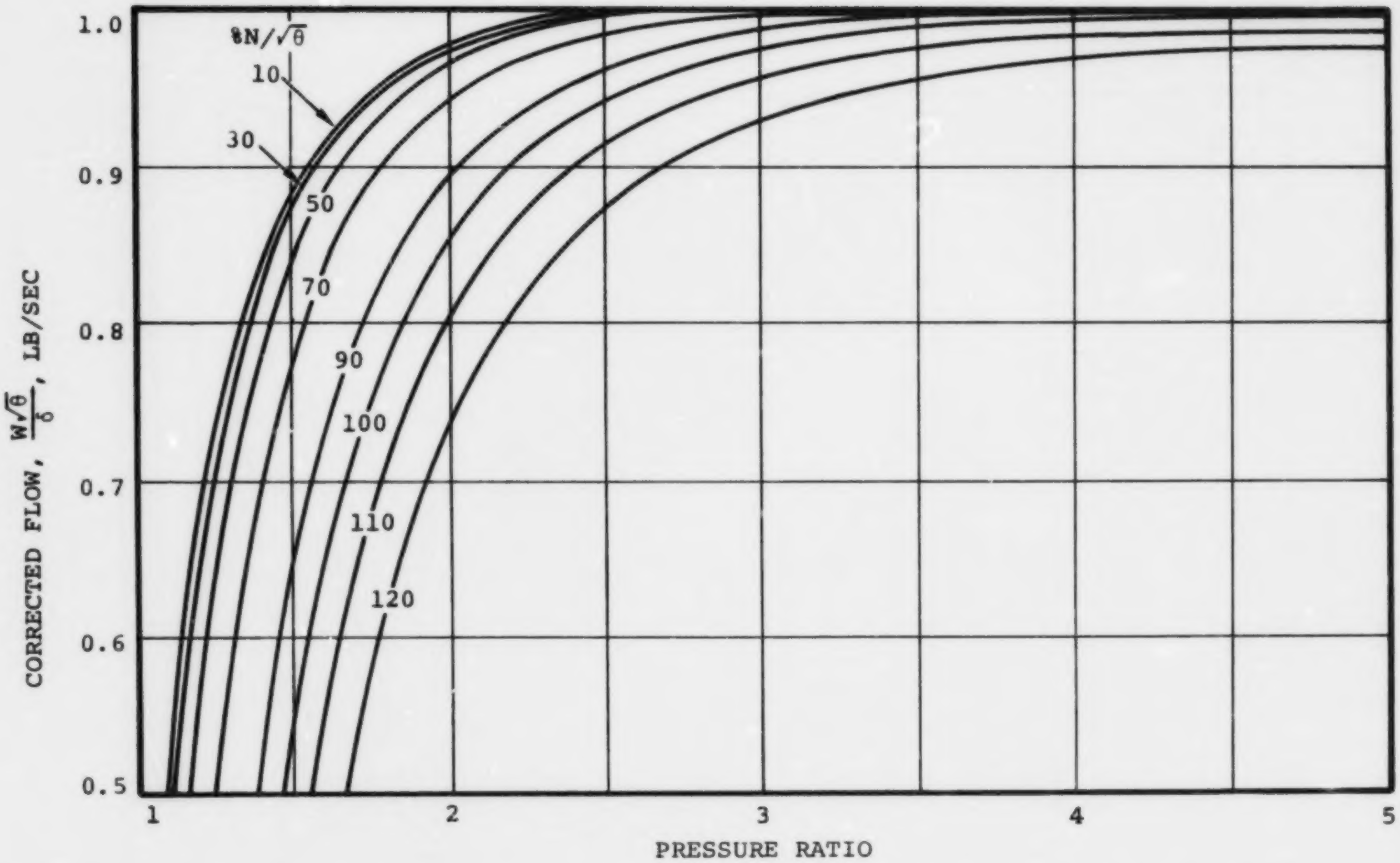
SINGLE-SHAFT REGENERATED ENGINE (NIII2V) TURBINE PERFORMANCE

FIGURE 5-10



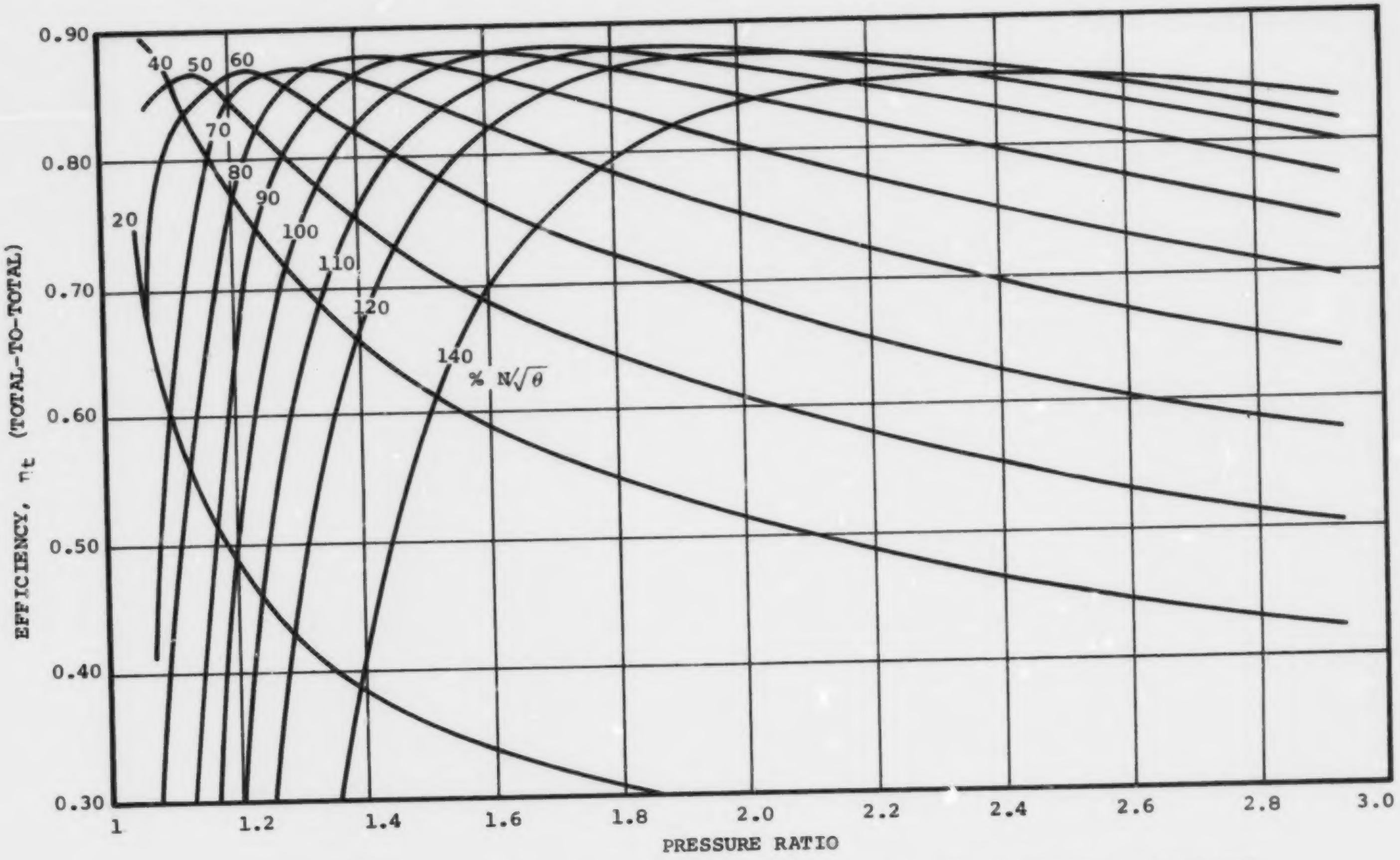
FREE-TURBINE REGENERATED ENGINE (AIII2V) GASIFIER TURBINE PERFORMANCE

FIGURE 5-11



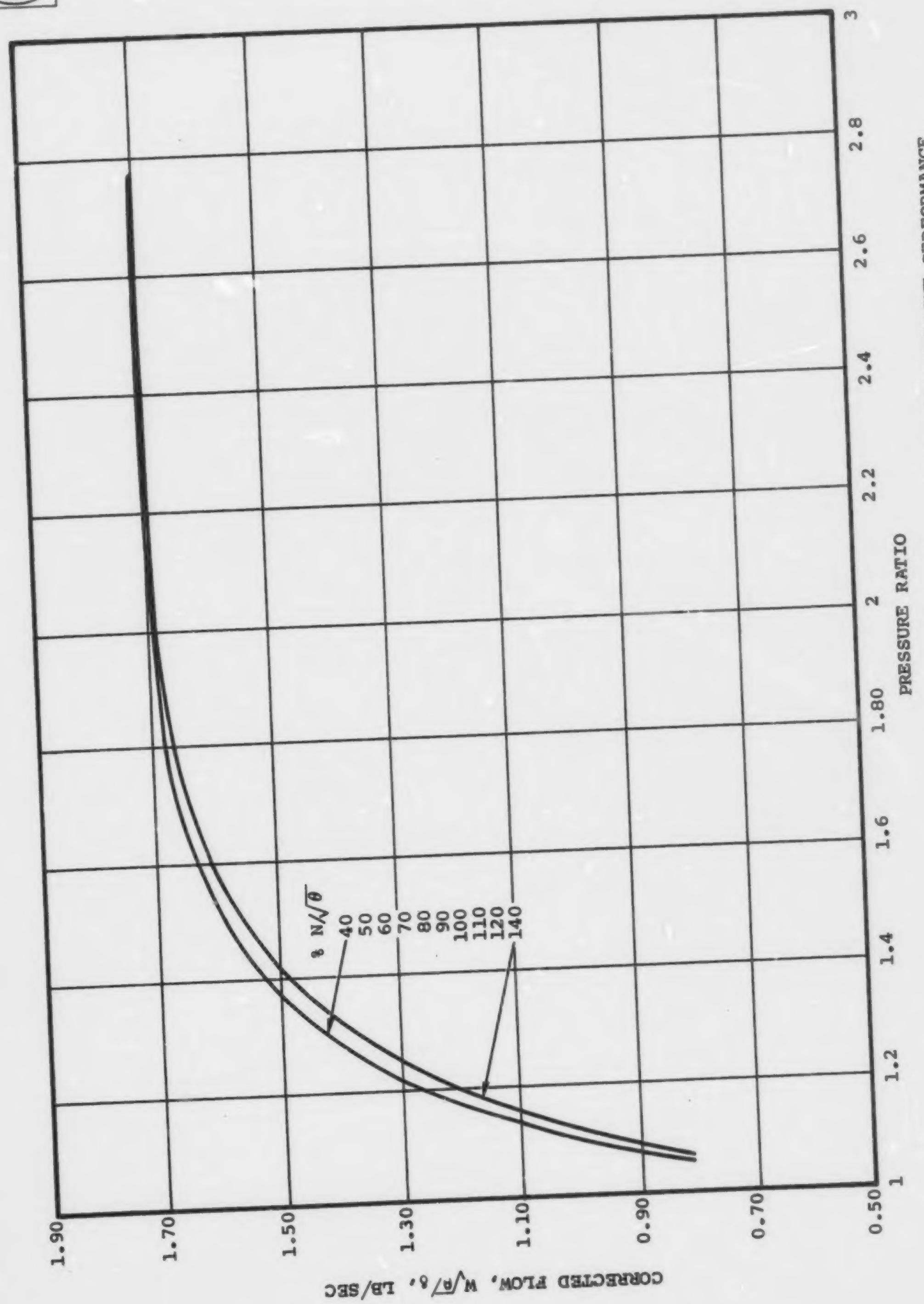
FREE-TURBINE REGENERATED ENGINE (AIII2V) GASIFIER TURBINE PERFORMANCE

FIGURE 5-12



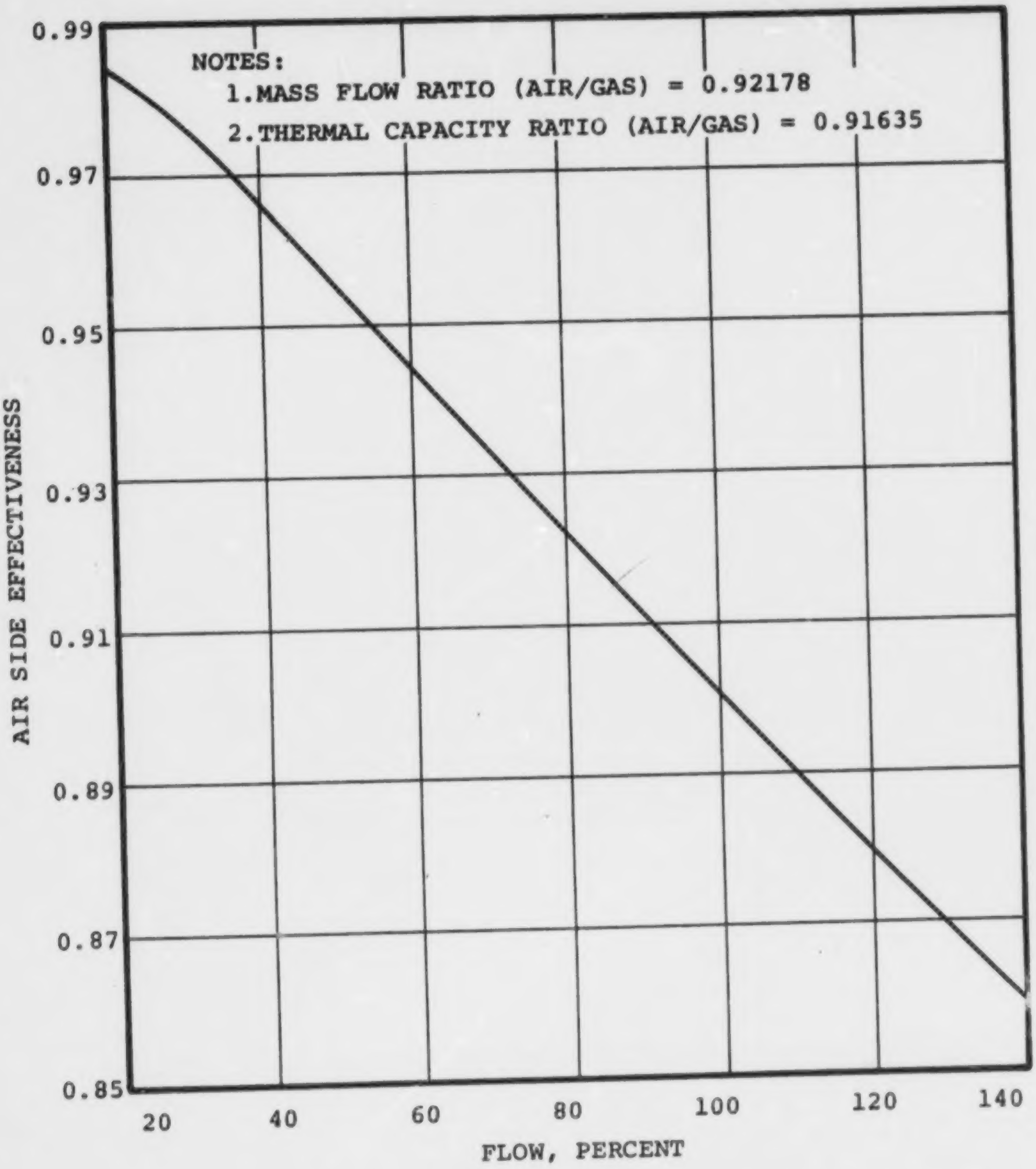
FREE-TURBINE REGENERATED ENGINE (AIII2V), $A_N/A_{ND} = 1.0$, POWER TURBINE PERFORMANCE

FIGURE 5-13



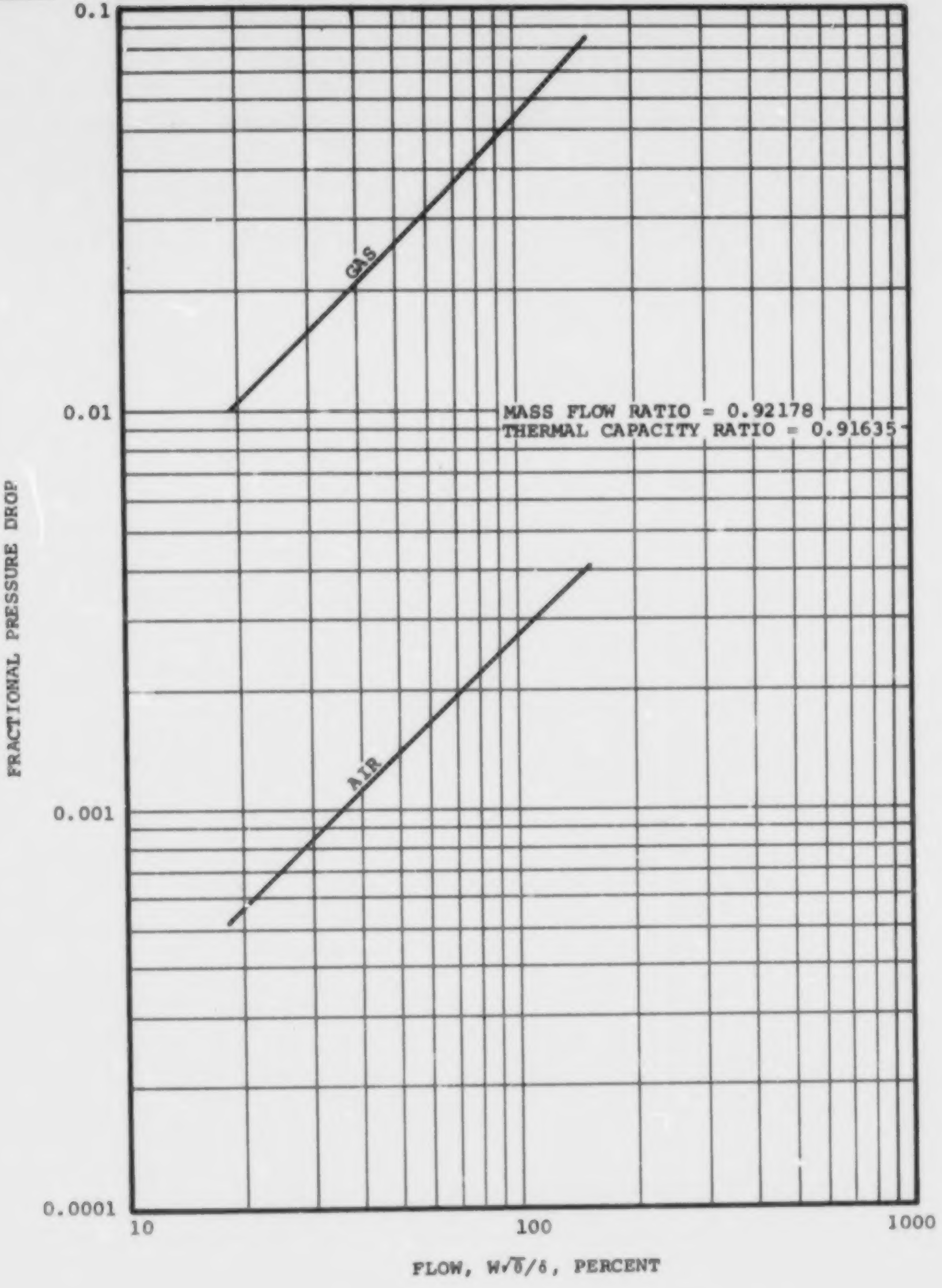
FREE-TURBINE REGENERATED ENGINE (AIII2V), A_N/A_{ND} = 1.0, POWER TURBINE PERFORMANCE

FIGURE 5-14



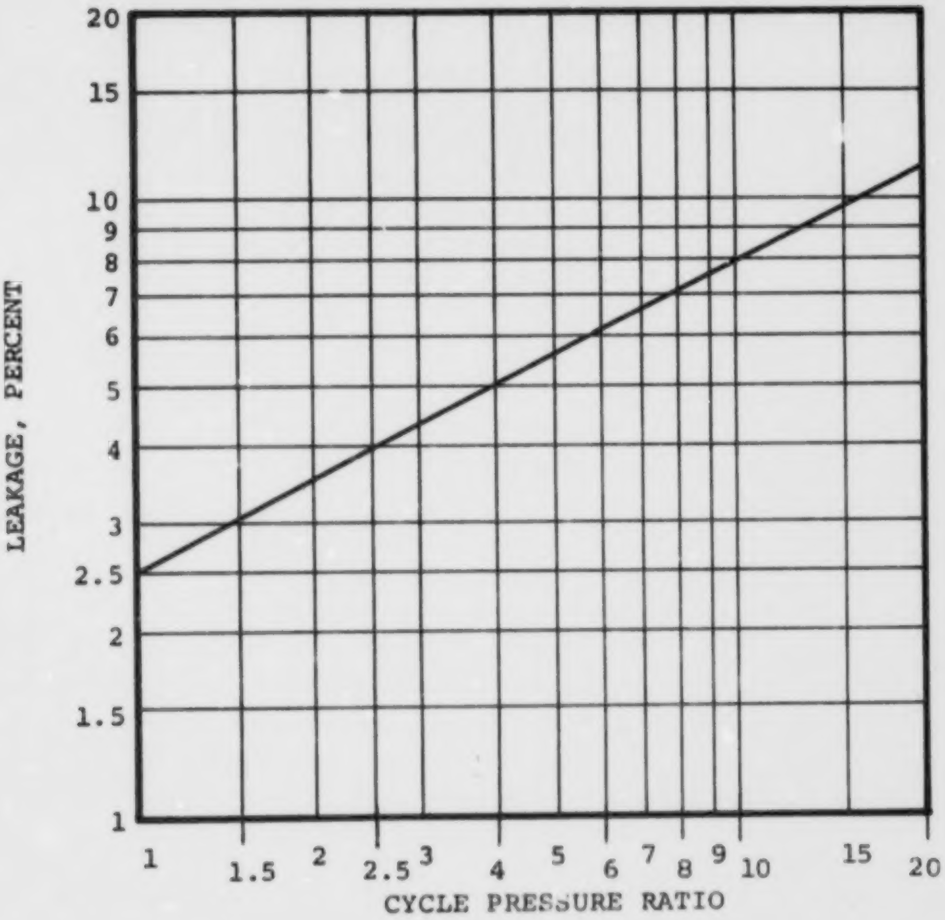
REGENERATOR OFF-DESIGN PERFORMANCE
(CERAMIC ROTARY TYPE)

FIGURE 5-15



ROTARY-CERAMIC REGENERATOR OFF-DESIGN PERFORMANCE

FIGURE 5-16



REGENERATOR SEAL LEAKAGE GOALS
AT CYCLE DESIGN PRESSURE RATIO

FIGURE 5-17

Fixed-boundary recuperator effectiveness as a function of mass flow is shown in Figure 5-18.

5.3 ENGINE PERFORMANCE MAPS

5.3.1 Recuperated, Single-Shaft Cycle

Engine performance maps for the recuperated, single-shaft cycle (NII2V) are shown in Figures 5-19 through 5-22 for 105°, 85°, 59°, and 30°F ambient temperatures, respectively.

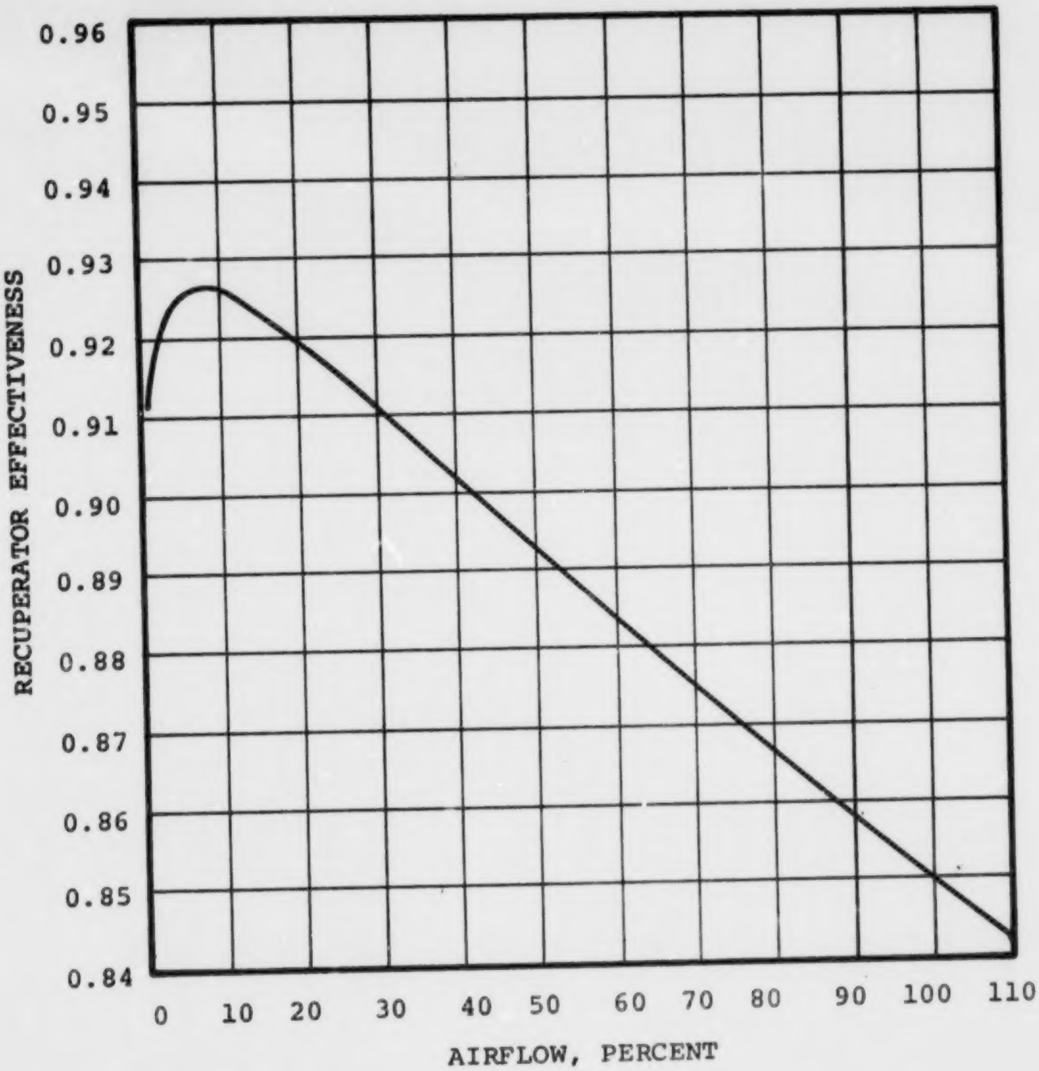
The performance shown is for an engine sized for a 108-shp, sea-level, standard-day rating. Performance and engine size was scaled in the mission analysis program to 125-shp rating to meet vehicle performance goals. A power augmentation ratio $[(\text{shp})_{\text{wet}}/(\text{shp})_{\text{dry}}]$, with water injection of 1.3, was required for maximum power operation on a 105°F day. Based on water-injection tests of the T-76 turboprop engine, the fuel flow-rate was also increased in proportion to the power augmentation ratio.

Compressor variable inlet guide vanes are used to improve the specific fuel consumption by allowing the engine to operate at a constant 1700°F turbine inlet temperature at part-load. The engine performance is shown for guide vane settings yielding flow ratios of 0.6 to 1.025 for a given compressor speed.

The idle speed was selected as 60 percent of the design rated speed. This was a compromise between less fuel consumption at lower speeds and decreased engine acceleration time for higher idle speeds.

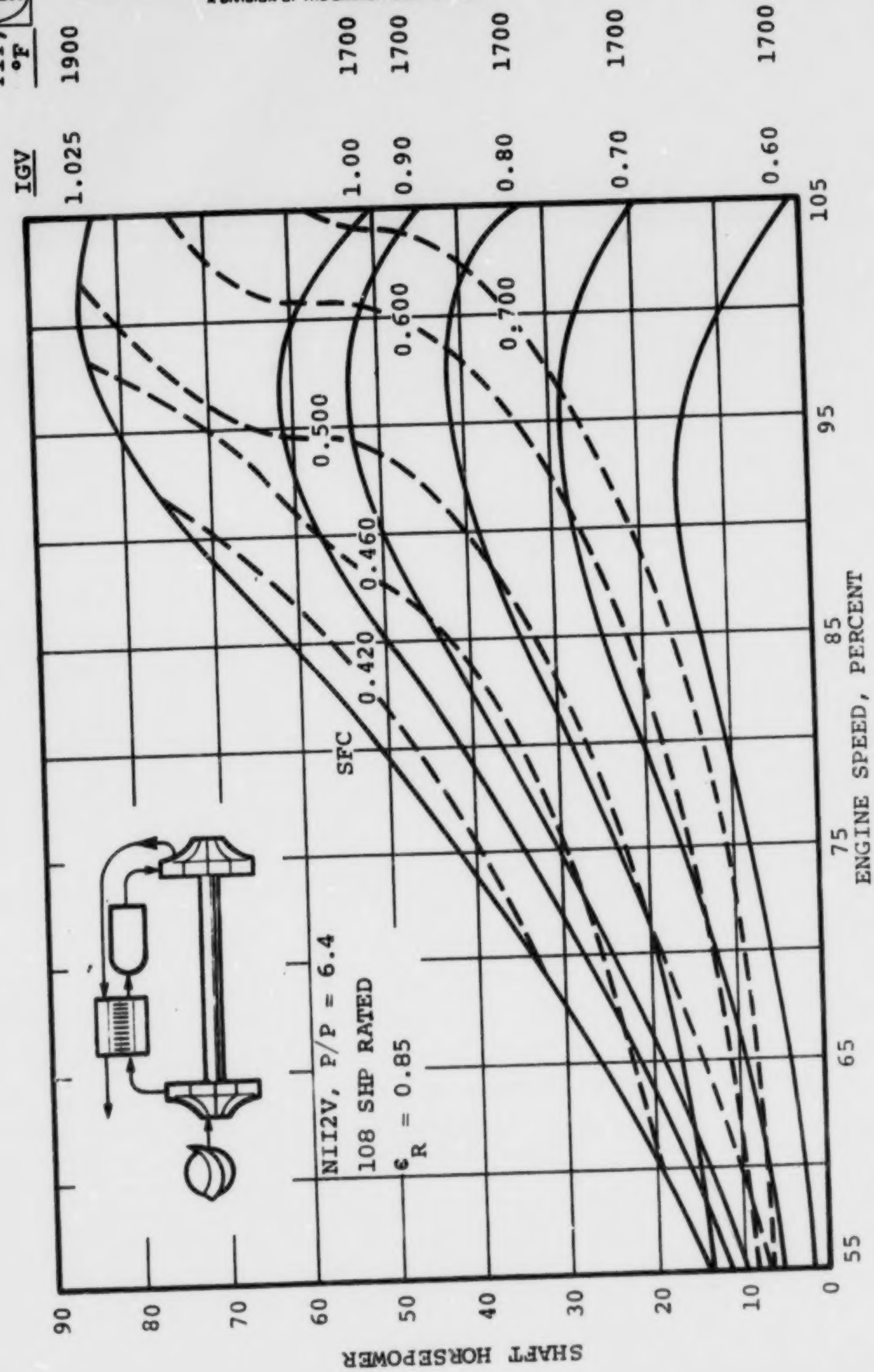
5.3.2 Regenerated, Single-Shaft Cycle

Performance maps, similar to the recuperated, single-shaft cycle, are shown in Figures 5-23 through 5-26 for the regenerated single-shaft



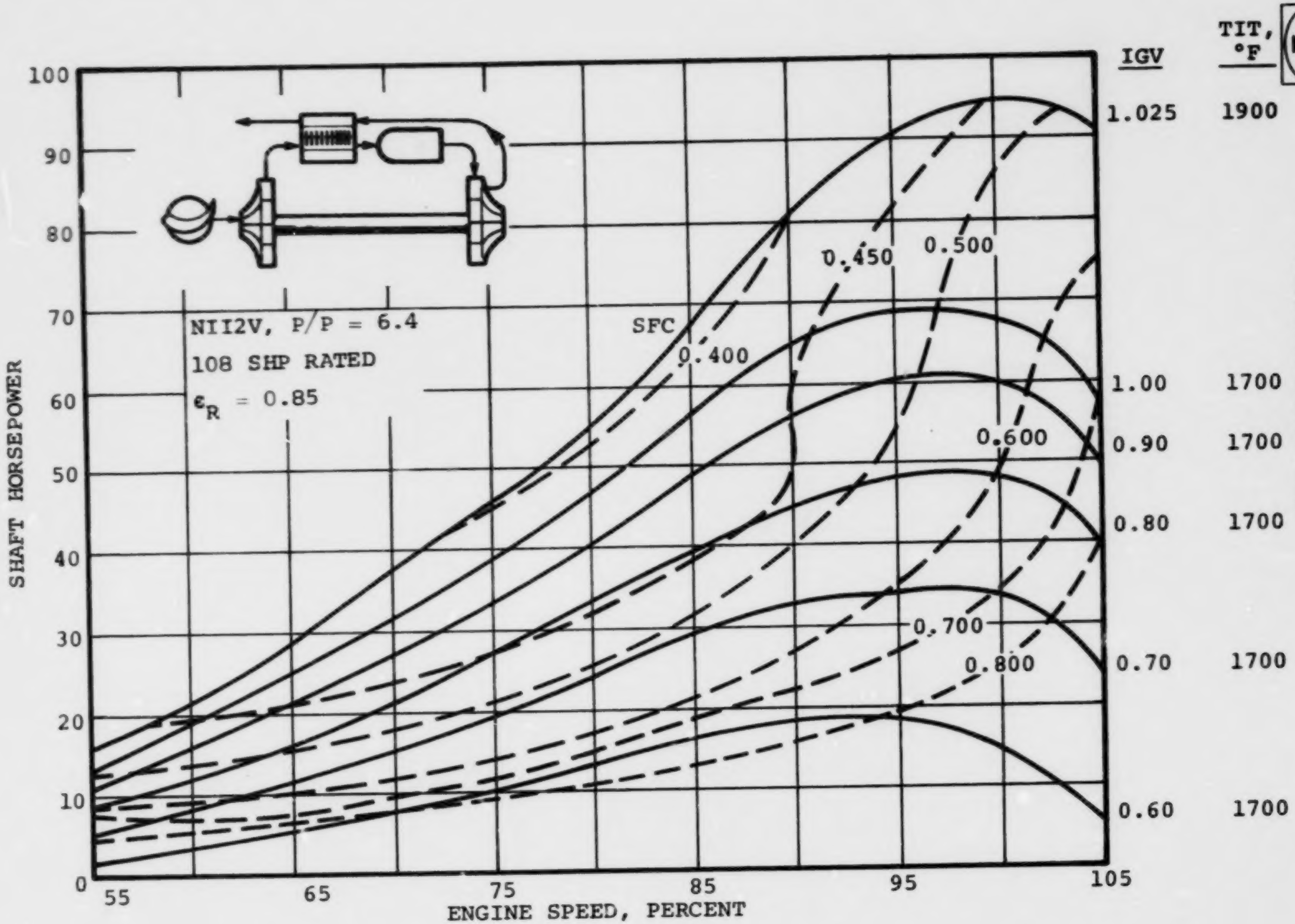
ESTIMATED METAL RECUPERATOR OFF-DESIGN PERFORMANCE

FIGURE 5-18



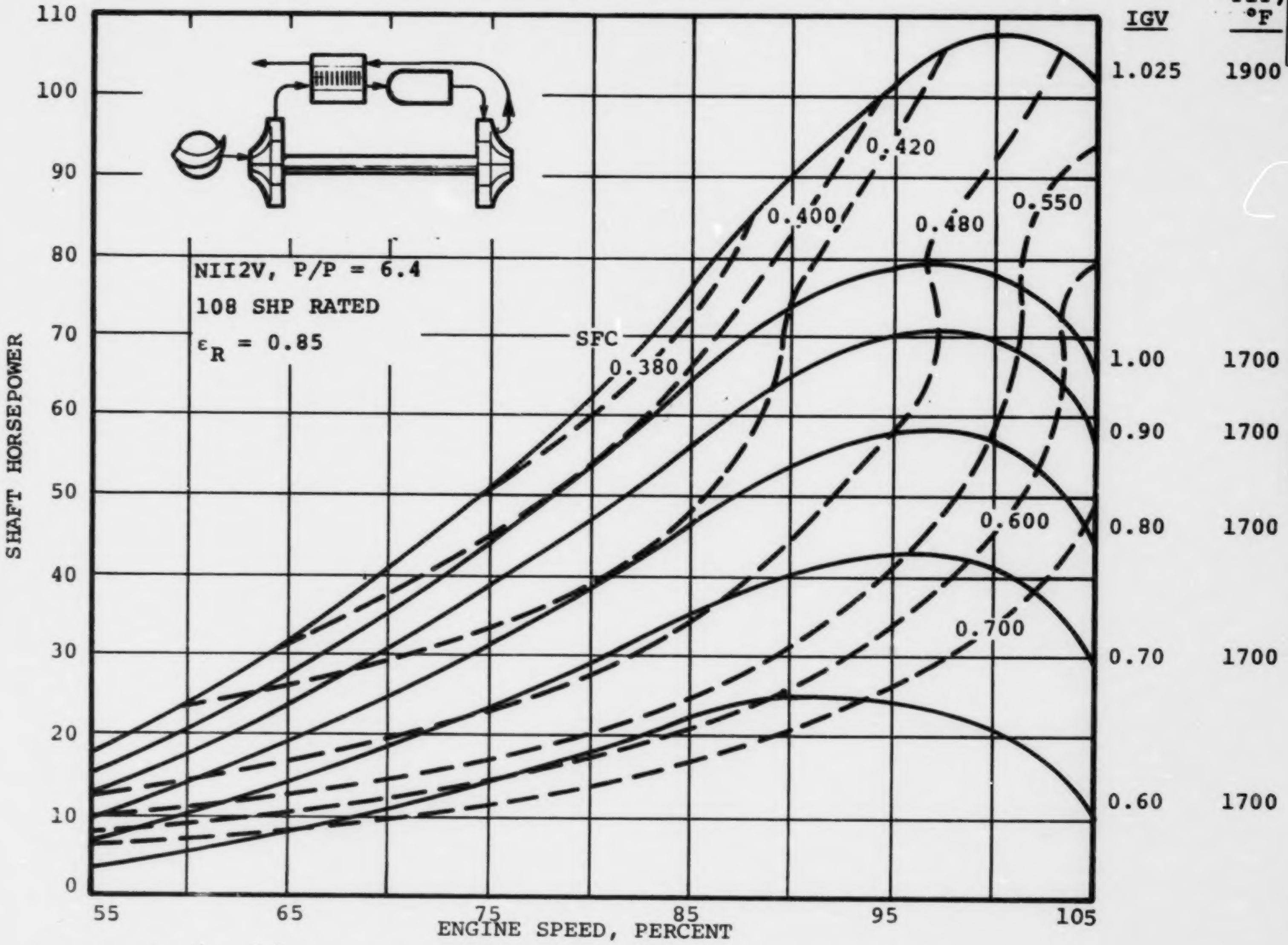
RECUPERATED, SINGLE-SHAFT CYCLE, 105°F DAY

FIGURE 5-19

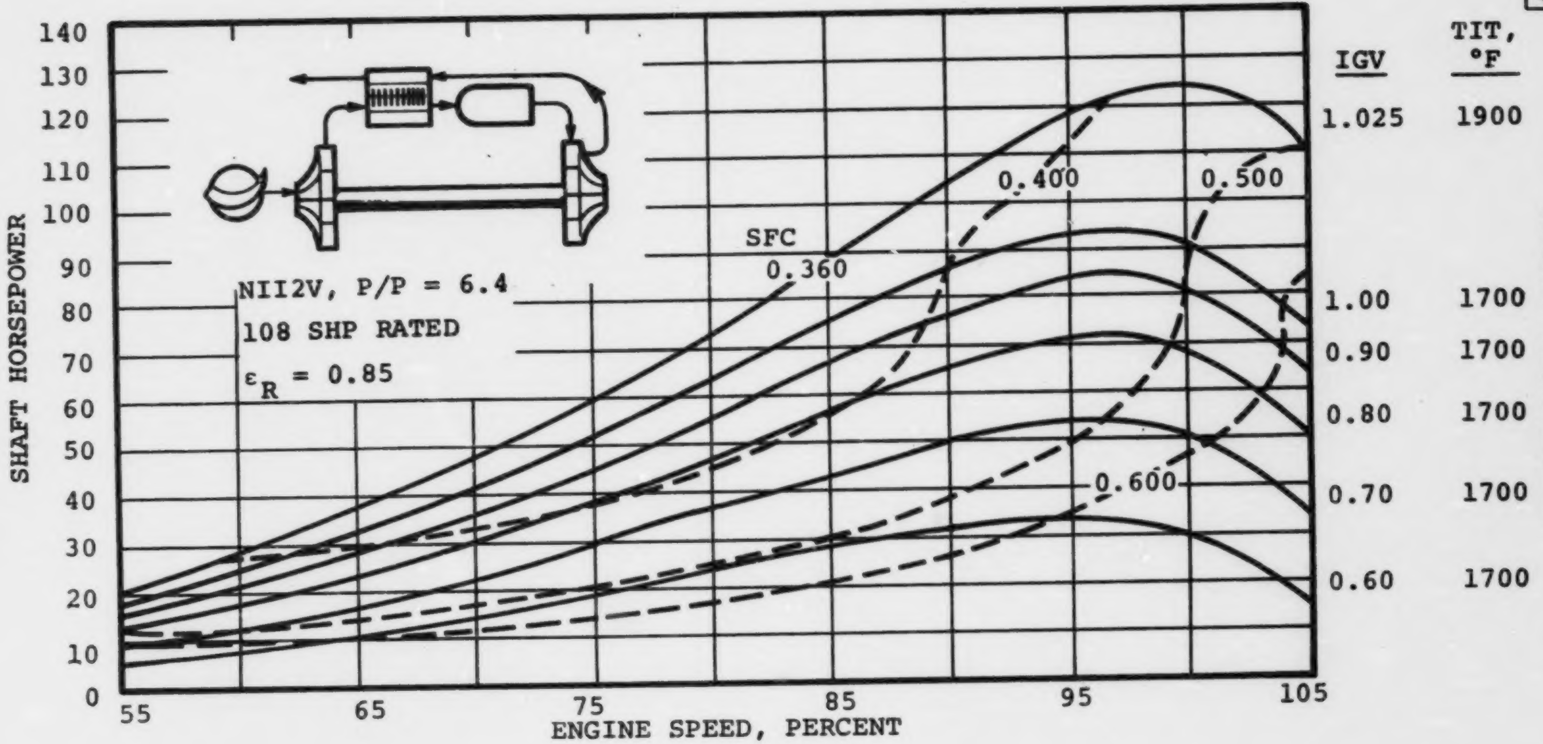


RECUPERATED, SINGLE-SHAFT CYCLE WITH VARIABLE INLET GUIDE VANES, 85°F DAY

FIGURE 5-20

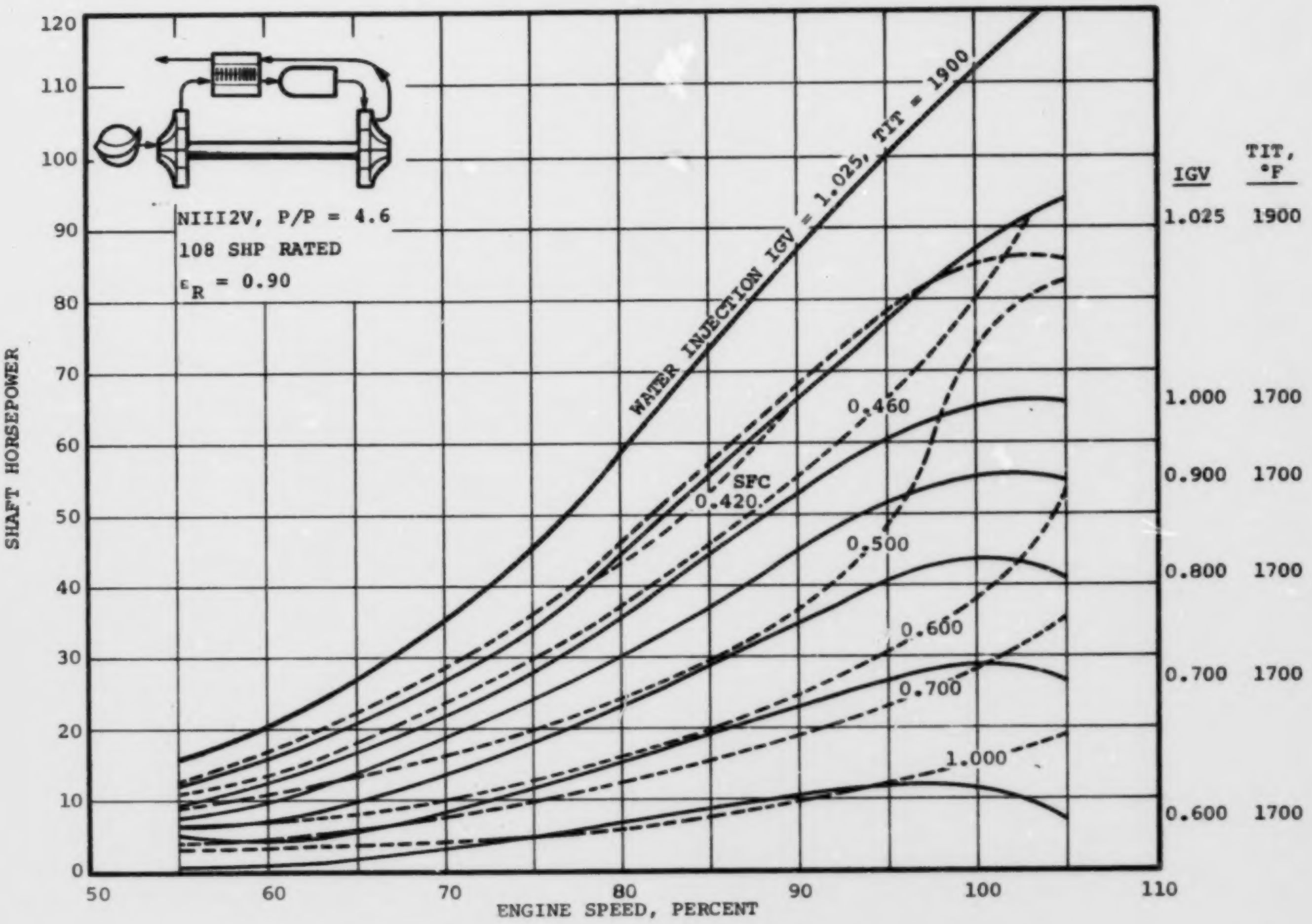


RECUPERATED, SINGLE-SHAFT CYCLE WITH VARIABLE INLET GUIDE VANES, 59°F DAY
FIGURE 5-21



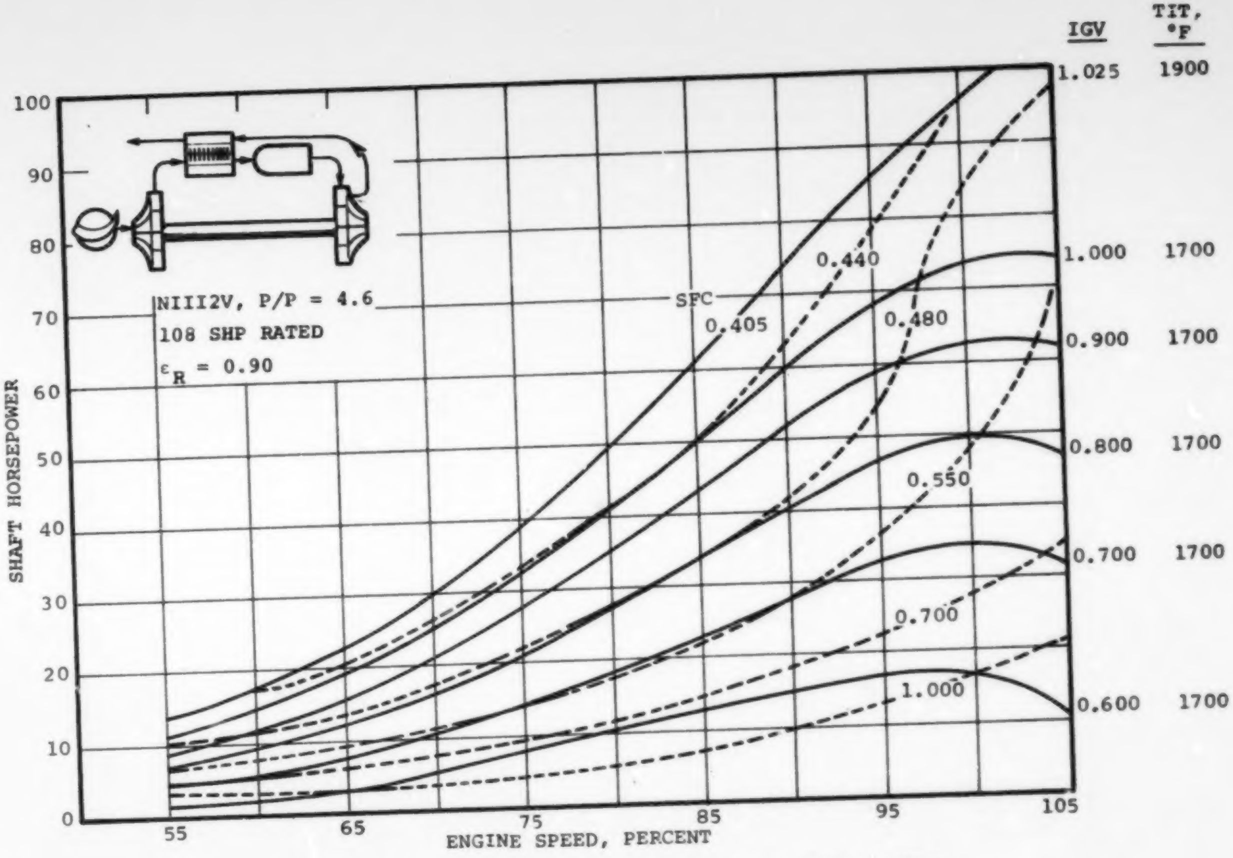
RECUPERATED, SINGLE-SHAFT CYCLE WITH VARIABLE INLET GUIDE VANES
30°F DAY

FIGURE 5-22



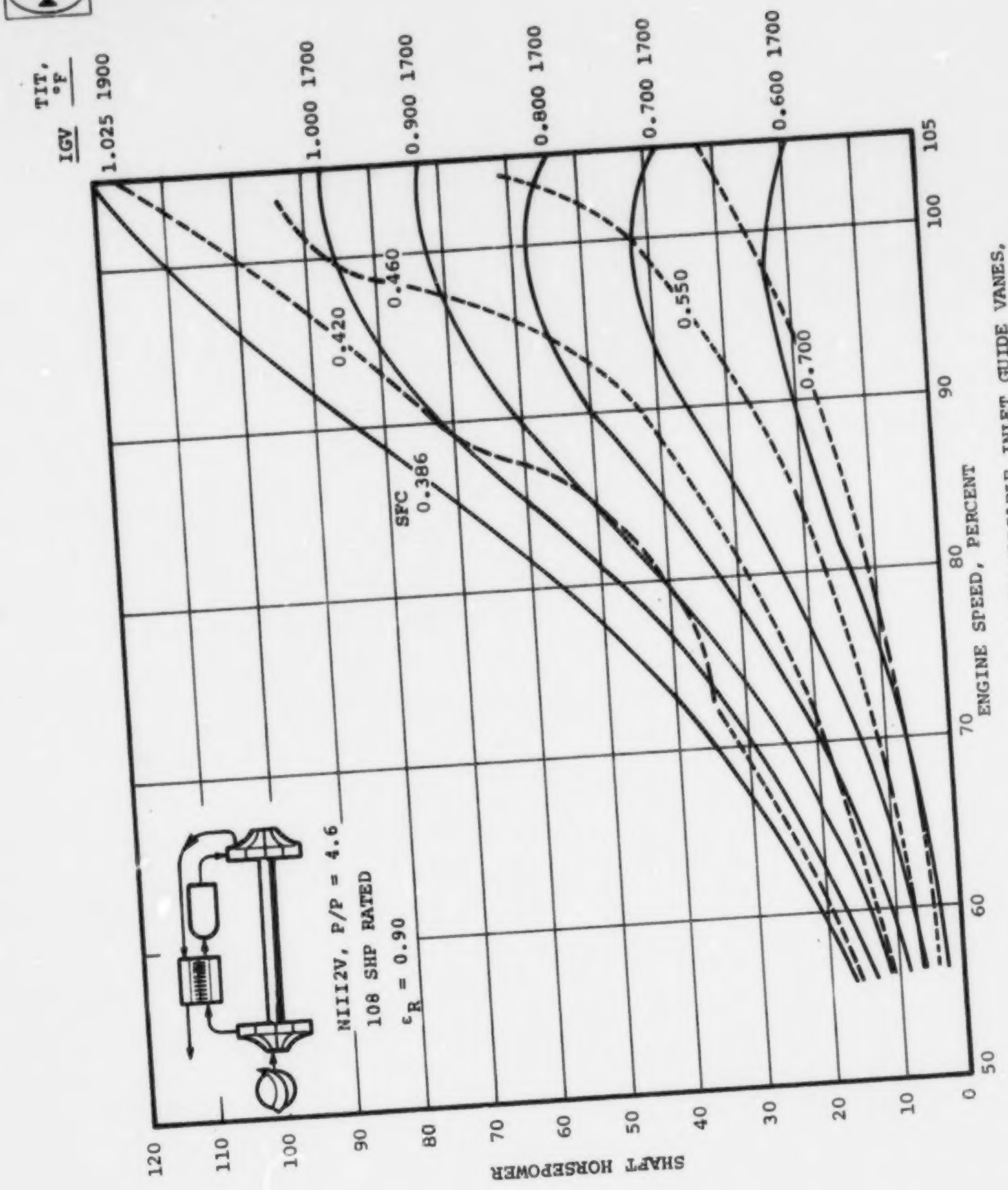
REGENERATED, SINGLE-SHAFT CYCLE WITH VARIABLE INLET GUIDE VANES
105°F DAY

FIGURE 5-23



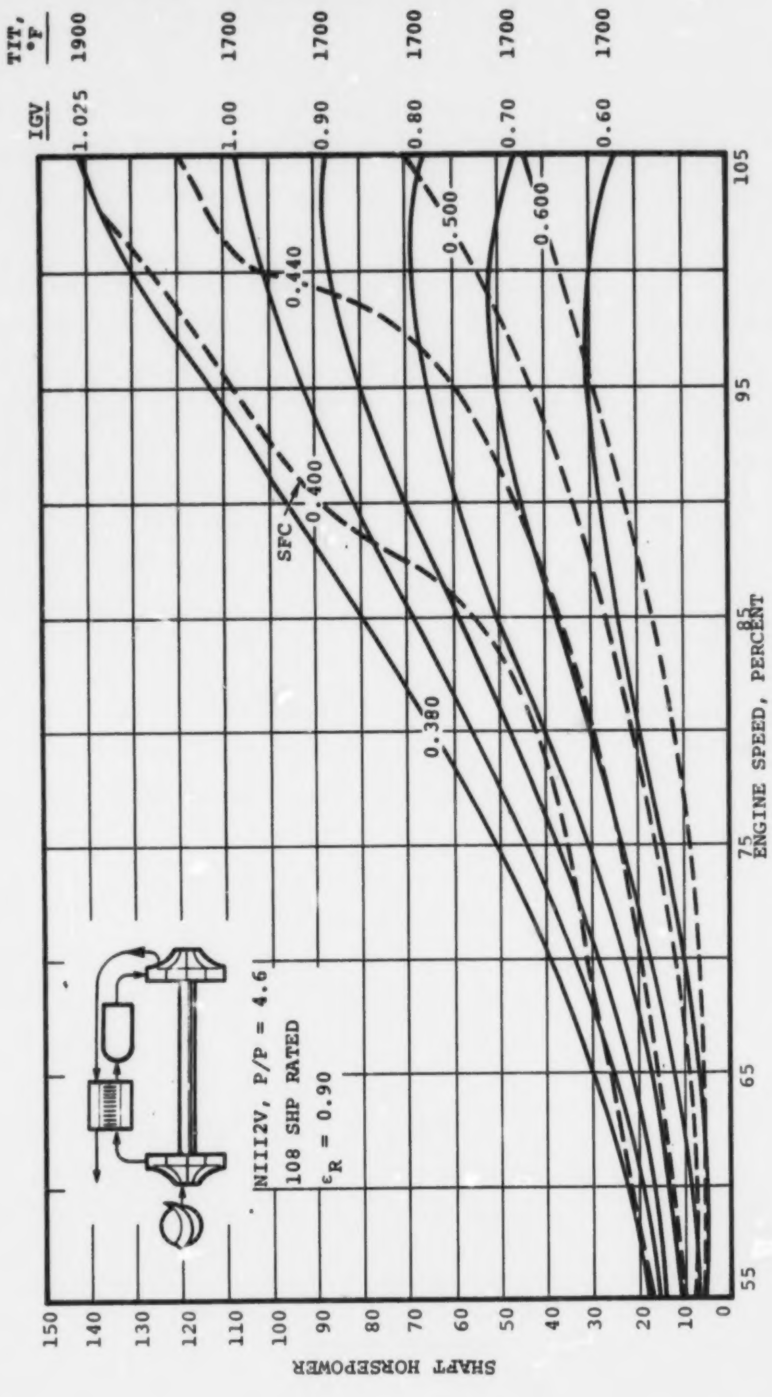
REGENERATED, SINGLE-SHAFT CYCLE WITH VARIABLE INLET GUIDE VANES
85°F DAY

FIGURE 5-24



REGENERATED, SINGLE-SHAFT CYCLE WITH VARIABLE INLET GUIDE VANES, 59°F DAY

FIGURE 5-25



REGENERATED, SINGLE-SHAFT CYCLE WITH VARIABLE INLET GUIDE VANES,
30°F DAY

FIGURE 5-26

cycle (NIII2V). The optimum cycle pressure ratio for the regenerated cycle is lower (4.6 versus 6.4), due to seal leakage and the higher effectiveness (90 versus 85 percent) of the rotary regenerator.

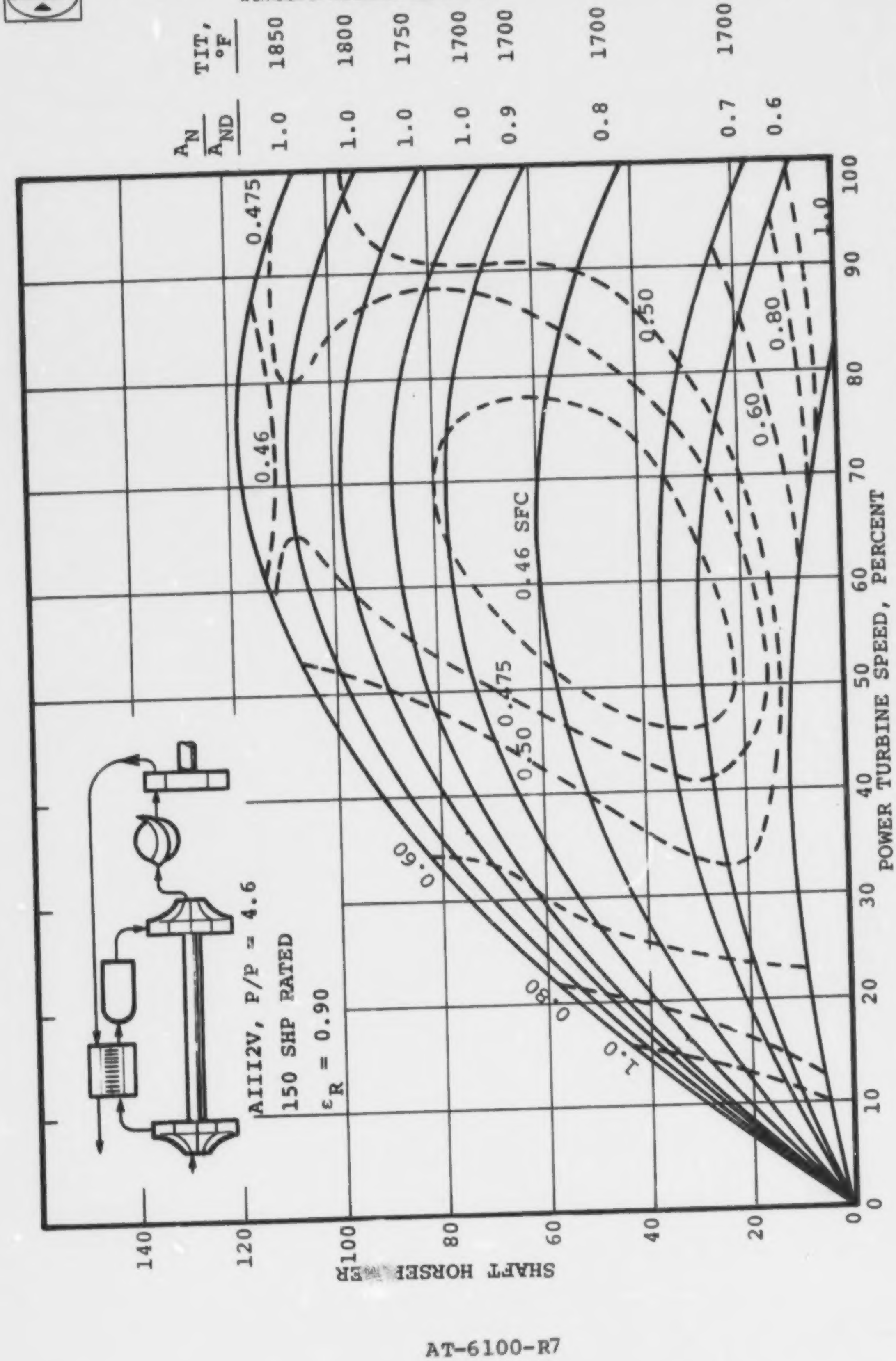
The sea-level, standard-day rated power, 108 shp, with a power augmentation ratio of 1.3, would successfully meet vehicle performance requirements, which were calculated for engines with and without power boost. An engine without power is required to have a rating of 155 shp on a standard day. The idle speed for both cycles was again chosen as 60 percent of the design rated speed--a compromise between engine acceleration and decreased fuel consumption.

5.3.3 Regenerated, Free-Turbine Cycle

Steady-state engine performance maps of the regenerated free-turbine cycle are shown in Figures 5-27 to 5-29 for ambient temperatures of 105°, 85°, and 59°F. The gas generator idle shaft speed is computed at approximately 48 percent of the design rated speed for these maps.

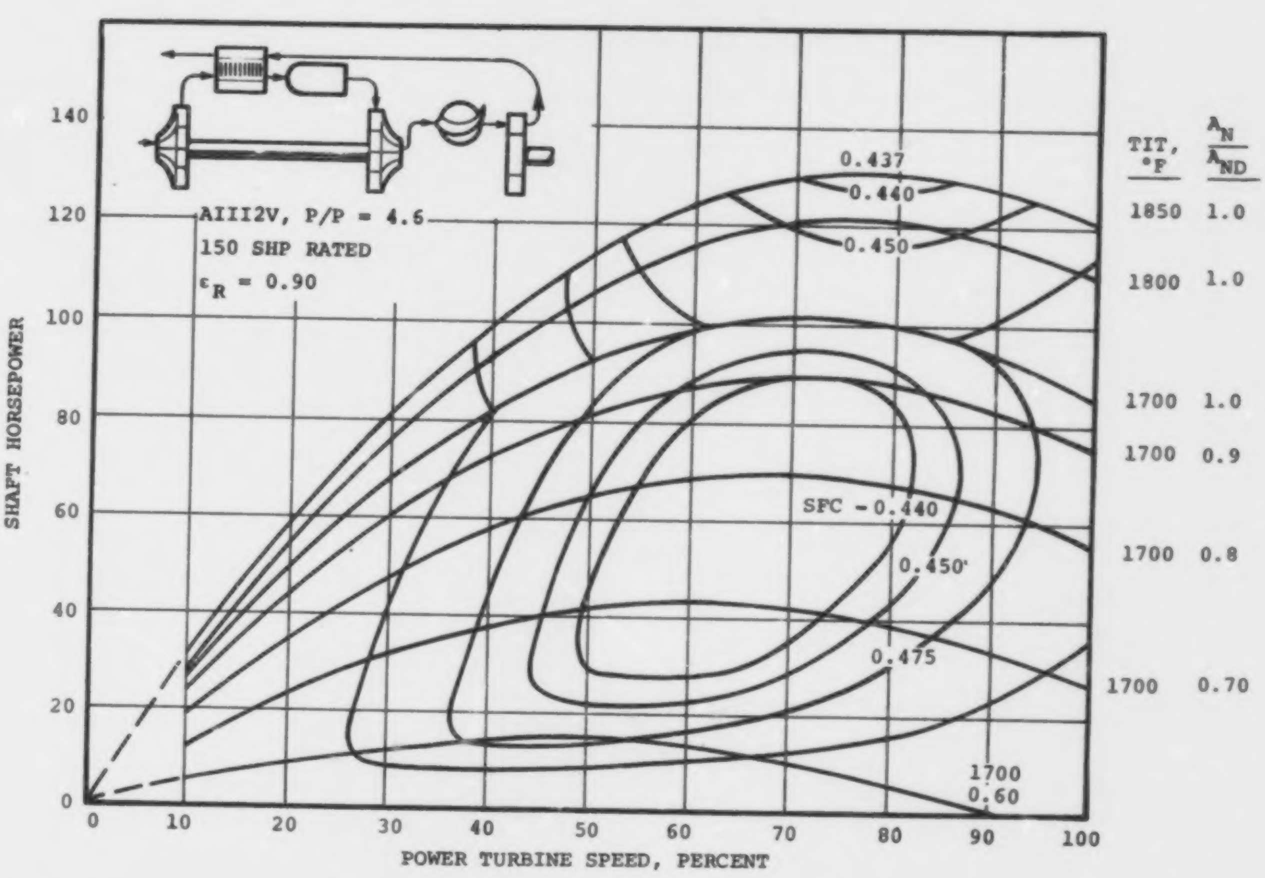
To account for the power and time required to accelerate the gas generator from one operating condition to another, transient performance maps were computed (Figures 5-30 to 5-32). The output shaft power, plus the power available for acceleration of the gas generator shaft, is shown as a function of both the gas generator turbine and power turbine speeds. Performance was calculated, using the maximum cycle temperature (1850°F), with the variable power turbine nozzles in an open position.

The performance shown is for an engine rated at 150 shp on a standard day. To meet the vehicle acceleration requirements, the engine was scaled to 175 shp, and the gas generator idle speed was limited to 73 percent of design rated speed. Consequently, compressor inlet guide vanes were added to the cycle to improve idle fuel consumption.



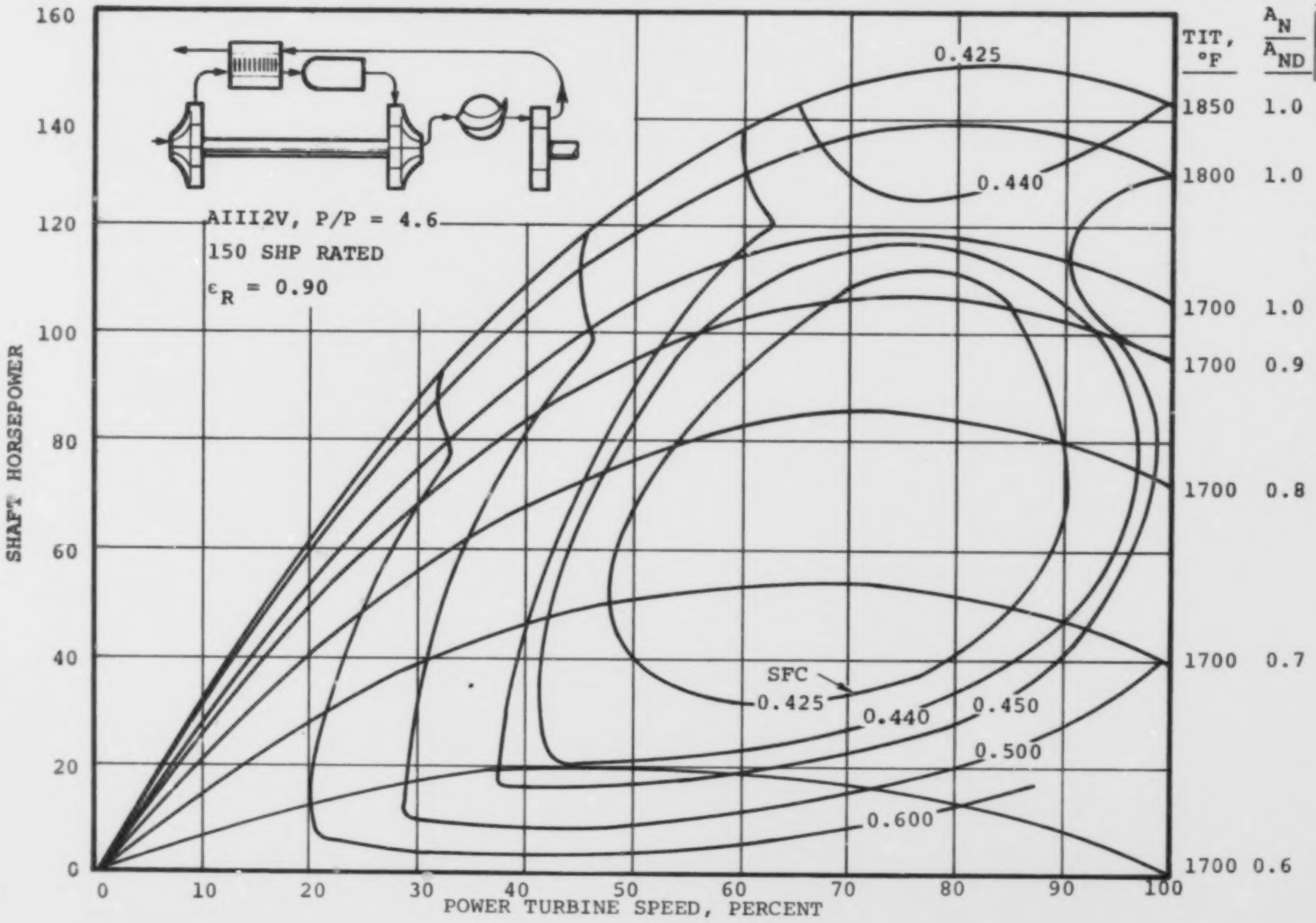
REGENERATED FREE-TURBINE WITH VARIABLE POWER TURBINE NOZZLES, 105°F DAY

FIGURE 5-27

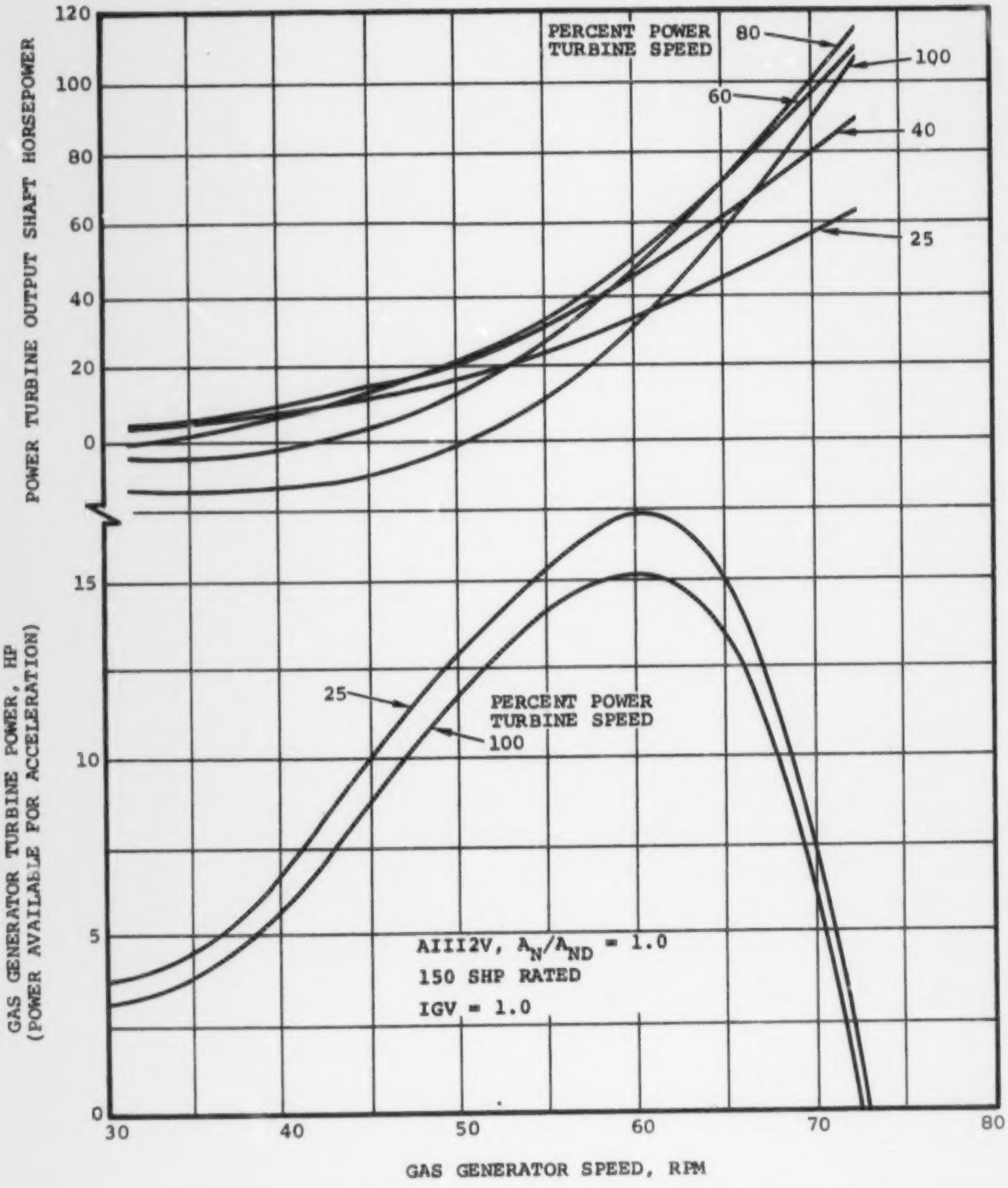


REGENERATED FREE-TURBINE WITH VARIABLE POWER TURBINE NOZZLES
85°F DAY

FIGURE 5-28

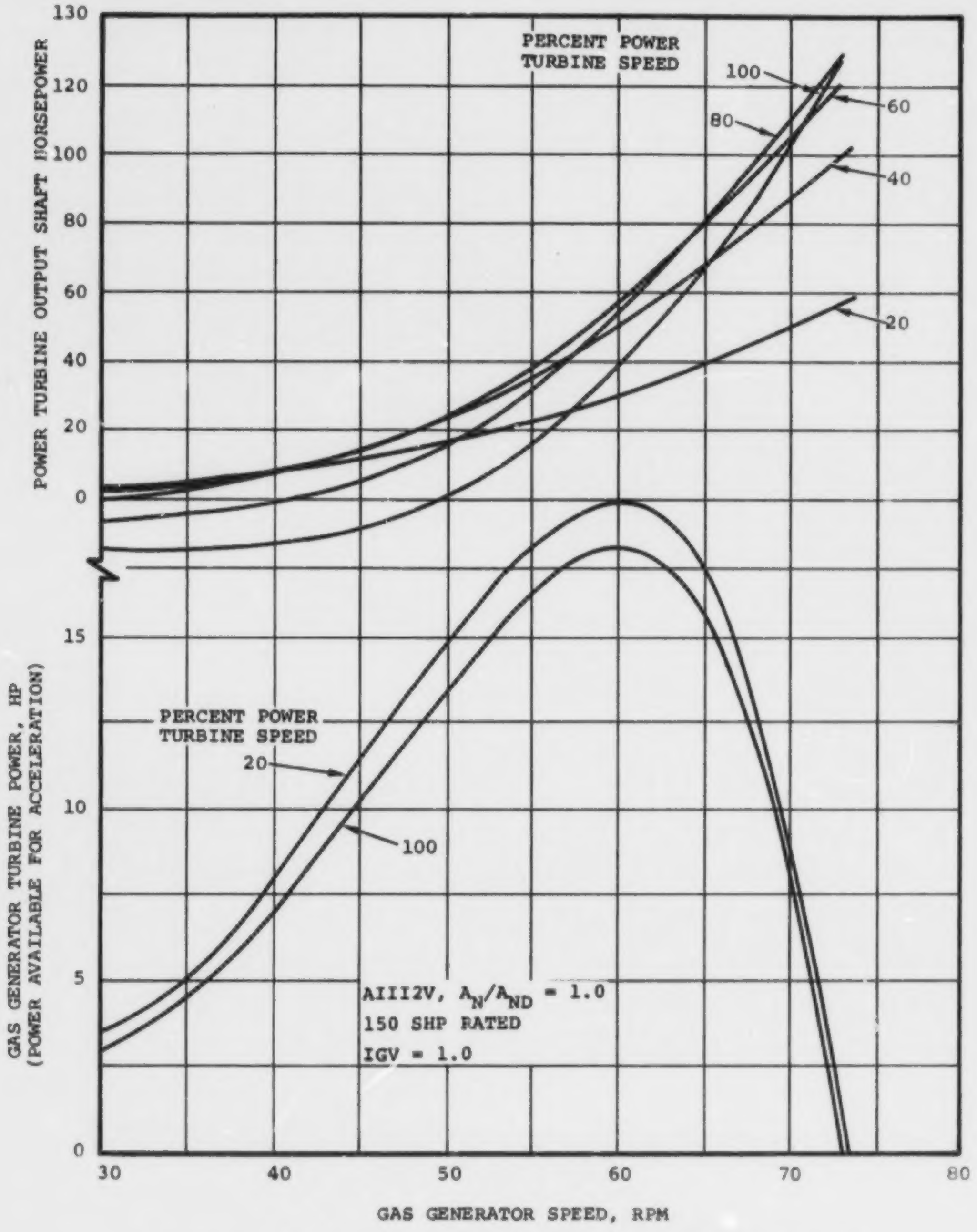


REGENERATED FREE-TURBINE WITH VARIABLE POWER TURBINE NOZZLES, 59°F DAY
FIGURE 5-29



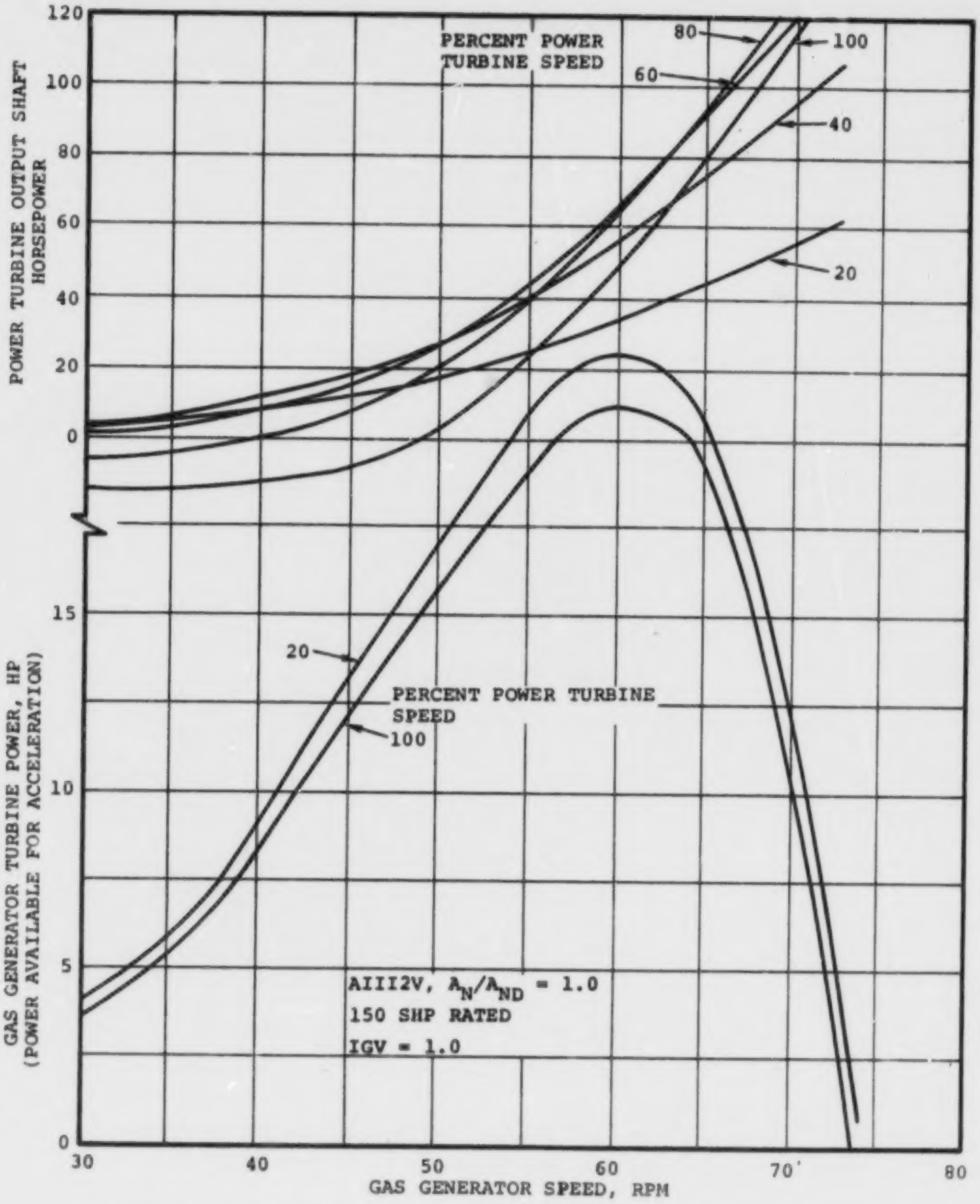
REGENERATED FREE-TURBINE ENGINE, 105°F DAY, UNBALANCED TORQUE

FIGURE 5-30



REGENERATED FREE-TURBINE ENGINE, 85°F DAY, UNBALANCED TORQUE

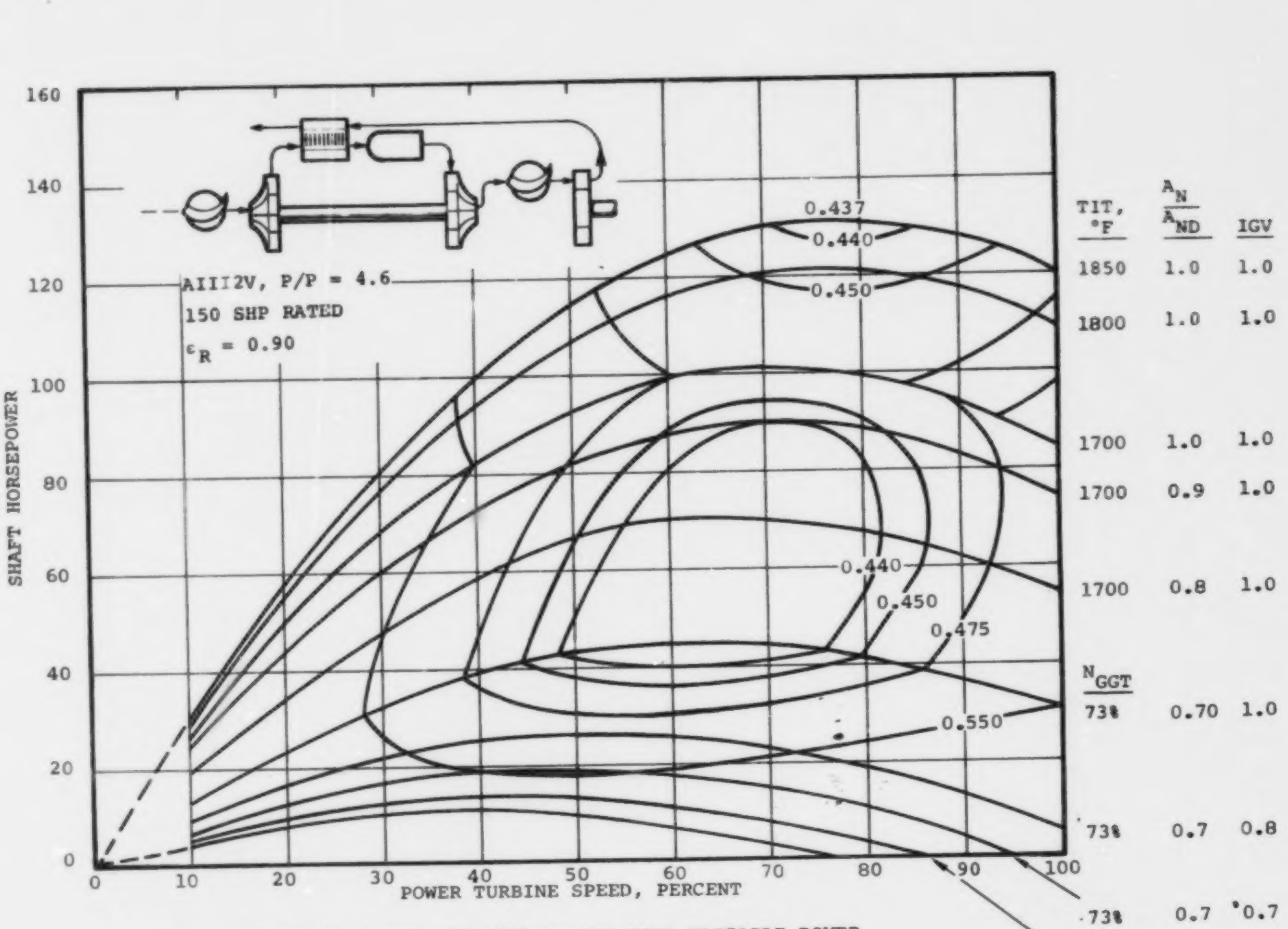
FIGURE 5-31



REGENERATED FREE-TURBINE ENGINE, 59°F DAY, UNBALANCED TORQUE

FIGURE 5-32

Since the requirement for inlet guide vanes was not known until late in the program, time did not permit recalculating engine performance for all ambient temperatures. Engine performance with inlet guide vanes for an 85°F day is shown in Figure 5-33. Transient performance would not be changed, because the guide vane would be in an open position (IGV = 1.0) during acceleration.



REGENERATED FREE-TURBINE ENGINE WITH VARIABLE POWER TURBINE NOZZLES AND VARIABLE INLET GUIDE VANES, 85°F DAY

FIGURE 5-33

SECTION 6

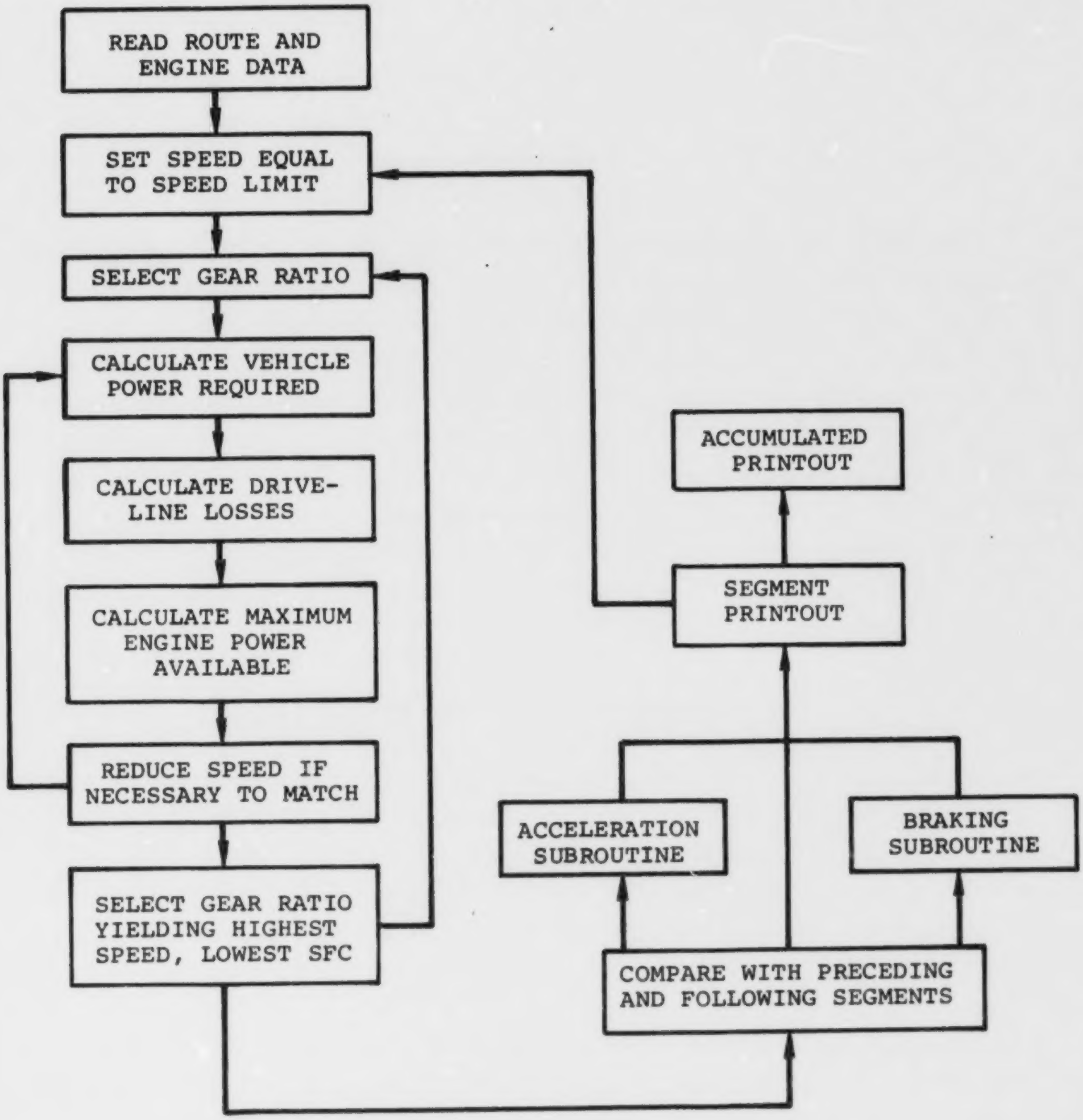
MISSION ANALYSIS

6.1 COMPUTER PROGRAM

A mission analysis computer program, formulated during a previous company-sponsored effort, was modified for the automobile gas turbine optimization study. The program matched the engine characteristics to the route profile, with due consideration for engine operating speed range, varying transmission efficiency, and vehicle tire adhesion limits. Two versions of the program--one for a single-shaft engine and the other for a free-turbine type--were created to facilitate mathematical modeling of engine dynamic characteristics. The program versions were sufficiently flexible to subject each candidate engine to the various performance tests and driving cycles specified for this study.

A block diagram of the mission analysis program applicable to both types of engines is depicted in Figure 6-1. Route profile data was introduced by means of cards designating length, grade, and speed limit for individual route segments. In this manner, an actual trip could be simulated with a high degree of accuracy, providing a sufficient number of route subdivisions were used. The required computation time was a fraction of both the number of segments and the number of accelerations.

Engine map data consisted of net engine horsepower, fuel flow, gas generator and power turbine speed (where applicable), turbine inlet temperature, and emission indexes (gm/kg fuel) as a function of output speed and throttle setting. Net engine horsepower was defined as gross power output minus the load imposed by engine accessories and the engine-mounted speed reduction gearbox. A separate set of engine


READ ROUTE AND ENGINE DATA
SET SPEED EQUAL TO SPEED LIMIT
SELECT GEAR RATIO
CALCULATE VEHICLE POWER REQUIRED
CALCULATE DRIVE-LINE LOSSES
CALCULATE MAXIMUM ENGINE POWER AVAILABLE
REDUCE SPEED IF NECESSARY TO MATCH
SELECT GEAR RATIO YIELDING HIGHEST SPEED, LOWEST SFC
ACCUMULATED PRINTOUT
SEGMENT PRINTOUT
ACCELERATION SUBROUTINE
BRAKING SUBROUTINE
COMPARE WITH PRECEDING AND FOLLOWING SEGMENTS

MISSION ANALYSIS PROGRAM BLOCK DIAGRAM

FIGURE 6-1

map cards was required for each ambient temperature of interest. Fuel consumption for all gas turbine engines in this study was based on a specific gravity of 0.748 (6.25 lb/gal) and a lower heating value of 18,500 Btu/lb. Other input data included engine-to-transmission speed reduction, maximum vehicle speed at maximum engine speed, engine design speed, gas generator and power turbine inertia, and vehicle total weight.

A two-dimensional interpolation subroutine determined any desired engine parameter as a function of two other variables. Consequently, the engine was not restricted to a single speed/horsepower operating line but was free to seek the best match-point for a given load condition, in accordance with gear-shift logic built into the program.

The initial calculation for a given route segment involved identification of transmission gear ratios that would permit the speed limit to be achieved on a steady-state basis. If the engine power were insufficient to reach the speed limit, a lower match-point vehicle speed was determined by iteration. If it were not possible to attain the speed limit in any gear ratio, the gear yielding the highest car speed was selected. When the speed limit was attainable in more than one gear ratio, the ratio was chosen that provides the optimum fuel consumption. Infinitely variable transmissions were simulated by means of a large number of discrete gear splits.

Engine and vehicle inertia were ignored when a computer run was designated as steady-state on the appropriate control card. If such were the case, the calculation proceeded from segment to segment, as described above, until the route was completed. Individual segment printout information included length, grade, speed limit, required engine horsepower, actual vehicle speed, gear, turbine inlet temperature, power turbine speed, gas generator speed, torque converter efficiency, drive-line efficiency, segment time, fuel economy, cumulative distance and time, weight of fuel consumed, grams of individual pollutants emitted, and Btu per mile. The hydrocarbon and NO_X constituents were expressed as equivalent $CH_{1.85}$ and NO_2, respectively.

Segment printouts were optional and could be restricted to a few segments or eliminated entirely. An accumulated data printout containing the following information was displayed after completion of a run:

(a) Total number of segments

(b) Total distance, miles

(c) Total fuel consumed, lb

(d) Elapsed time, min

(e) Average fuel consumption, mpg

(f) Average speed, mph

(g) Number of gear changes

(h) Average fuel heat release, Btu/mile

(i) HC emission, gm/mile

(j) CO emission, gm/mile

(k) NO_X emission, gm/mile

An additional feature of the mission analysis program was an integrated operating time computation, applicable to power turbine speed, turbine inlet temperature, gas generator speed, engine horsepower, and other variables. This option allowed the operating time associated with designated brackets of speed, temperature, and power output to be computed and tabulated. The resulting data could be converted to histogram form to facilitate an examination of engine stress-life characteristics.

6.2 STEADY-STATE POWER

Power required to overcome rolling resistance, air drag, and grade resistance was computed, using numerical constants furnished by OAP,

HP_{ROL} = rolling resistance, horsepower

$$= \left(\frac{W \cos \theta}{65}\right)\left(\frac{V}{375}\right)\left[1 + 1.467\ V(R_1 + 1.467\ R_2 V)\right]$$

where:

V = vehicle test weight, lb

θ = grade angle

V = vehicle velocity, mph

R_1 = rolling resistance constant = 1.4×10^{-3}

R_2 = rolling resistance constant = 1.2×10^{-5}

$$HP_{WND} = \text{wind drag horsepower}$$

$$= (6.83 \times 10^{-6})\ \sigma \cdot C_D \cdot A_{fr} \cdot V^3$$

where:

$$\sigma = \frac{\text{air density}}{\text{density at 1 atm, 59°F}}$$

C_D = drag coefficient

A_{fr} = frontal area, ft^2

$C_C A_{fr}$ = 12

$$HP_{GRD} = \text{grade resistance horsepower}$$

$$= (W \sin \theta)\ \frac{V}{375}$$

A vehicle test weight of 4000 lb was used in most calculations, except those pertaining to the 30-percent grade performance requirement, where a test weight of 4700 lb was used. The 4000-lb weight was based on a preliminary estimate that the gas turbine engine would weigh about 300 lb less than the V-8 spark-ignition engine. The weights shown in Section 8 verify this preliminary estimate.

In accordance with OAP directives, the vehicle accessory power was set equal to 1.3 hp for the Federal Register route and 4.0 hp for all other routes.

6.3 REAR AXLE, TRANSMISSION, AND COUPLING CHARACTERISTICS

The drive-line horsepower losses associated with the rear axle were calculated as:

$$HP_{loss} = \frac{100 - A}{100} (HP_{in}) + f(N)$$

where:

A = torque efficiency, percent

HP_{in} = shaft power input

N = transmission input speed or rear axle output (wheel) speed

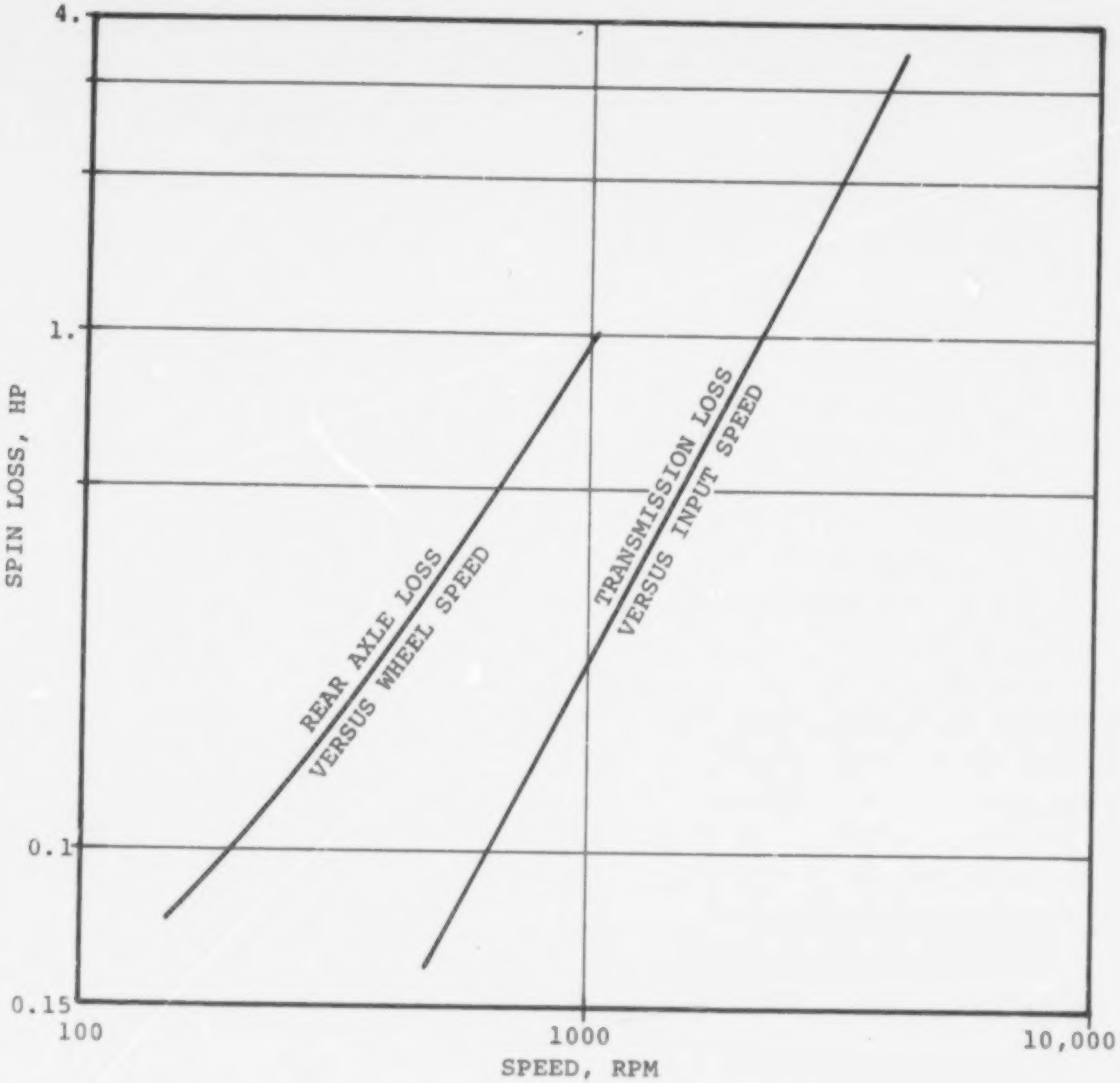
$f(N)$ = spin loss (includes windage and seal losses)

This procedure was also used for the conventional three-speed automatic transmission.

Torque efficiency was assumed independent of speed and power, while the spin-loss was dependent on speed only. Based on OAP data, the following torque efficiencies were assumed for the drive-line components of a typical medium-size passenger car with a three-speed automatic transmission:

Drive-Line Component	Gear Ratio	Torque Efficiency, %
Transmission (three-speed automatic)	2.5	95.6
	1.5	94.0
	1.0	100.0
Rear Axle	3.22	96.0

Transmission and rear axle spin-loss are plotted versus shaft speed in Figure 6-2. Numerical constants for the transmission spin-loss equation have been adjusted to yield the best correlation with



REAR AXLE AND TRANSMISSION SPIN-LOSS CHARACTERISTICS
(BASED ON OAP DATA)

FIGURE 6-2

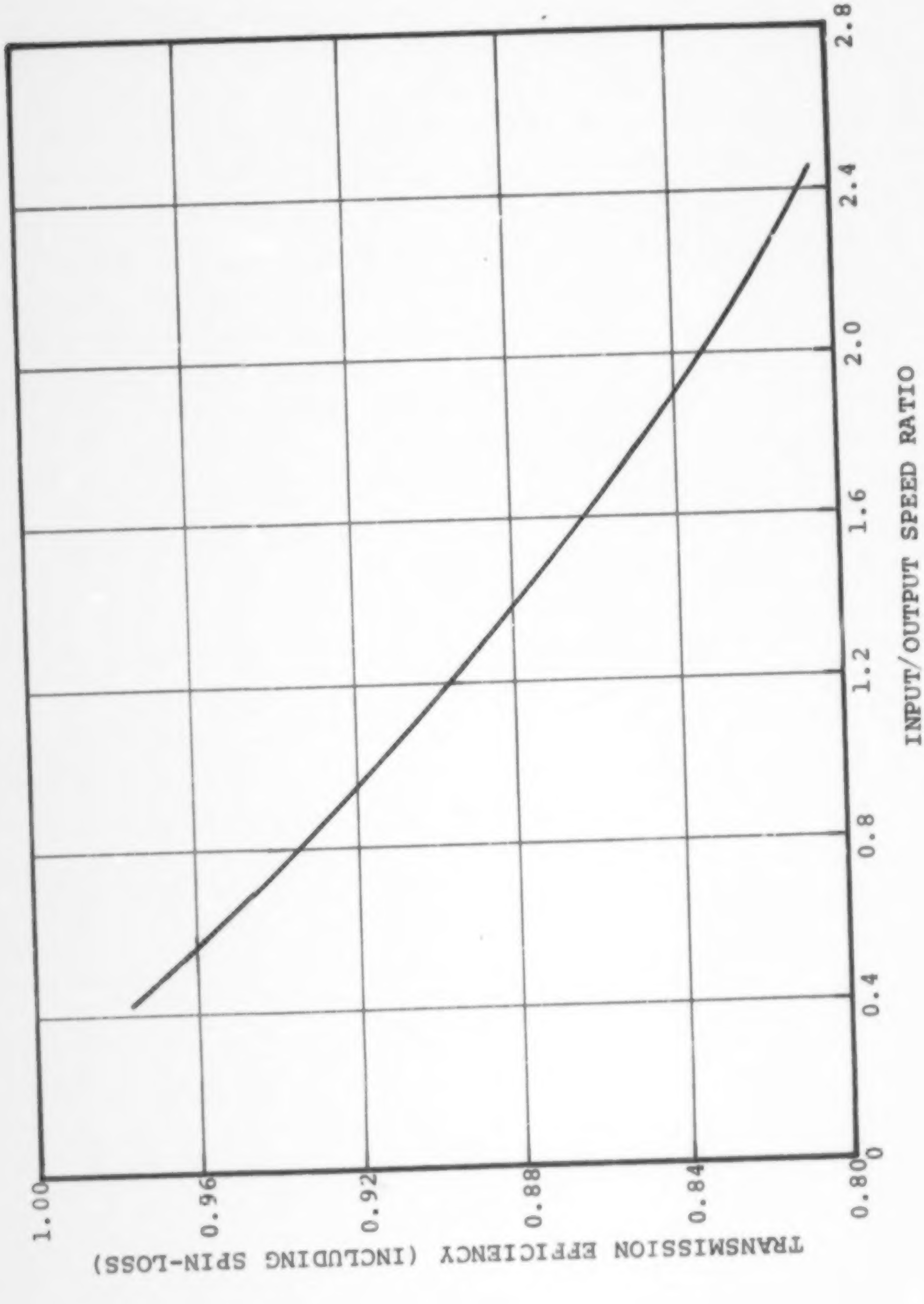
experimental data at the lower gear ratio, where the spin losses are especially significant.

Identical torque and spin-loss characteristics were assumed for the free-turbine engine transmissions. Gear ratios of 3.0, 1.65, and 1.0 were selected, based on engine/load matching optimization relating to the free-turbine engines under investigation.

Since a free turbine functions as a pneumatic torque converter, a supplementary hydraulic torque converter was not needed in conjunction with the transmission described above. A controlled shipping clutch characteristic was assumed to permit an idle speed, yielding minimum fuel consumption. A fill-and-drain fluid coupling could accomplish the same objective, but a conventional fluid coupling would impose an excessive load on the engine during idle periods.

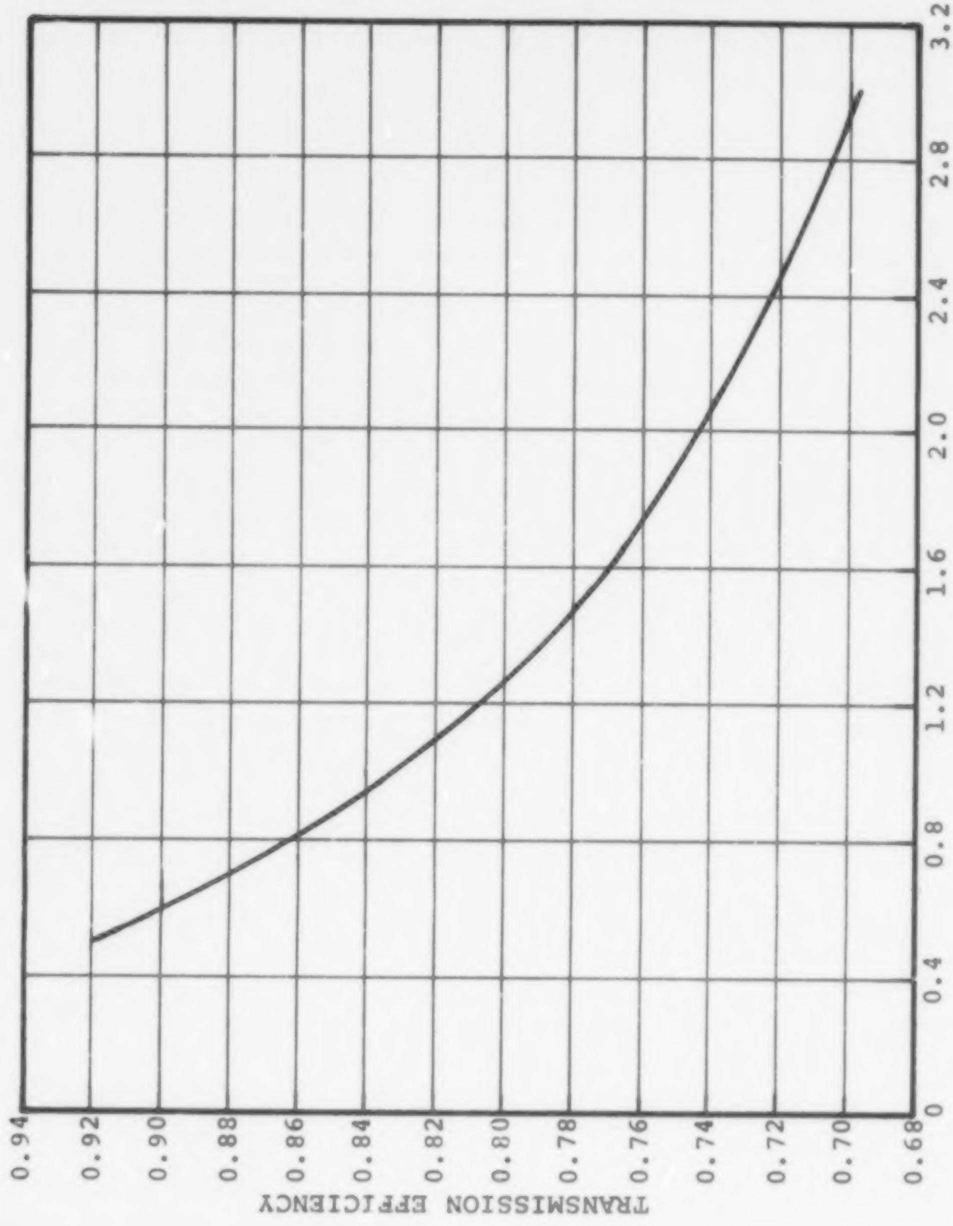
For single-shaft engines, an infinitely variable belt- or traction-drive transmission and slipping-clutch configuration was assumed for the mission analysis. Since a detailed breakdown of the losses was not available for this type of transmission, spin-loss was calculated with torque efficiency instead of separately. Belt-drive transmission efficiency is plotted versus input/output speed ratio in Figure 6-3, based on the nominal relationship between speed ratio and horsepower that would prevail for travel over a level road at constant speed. Rear axle losses are calculated as before. Similarly, traction-drive transmission efficiency is shown in Figure 6-4.

Since it was necessary to vary the speed ratio in discrete steps, the computer program was dimensioned to allow a maximum of 40 gear ratios. Vehicle performance, with an infinitely variable transmission could be closely approximated in this manner. The number of speed ratios was dictated by a compromise between accuracy and computer time.



BELT TRANSMISSION - ROAD-LOAD EFFICIENCY

FIGURE 6-3



TOROIDAL TRANSMISSION - ROAD-LOAD EFFICIENCY

FIGURE 6-4

The torque converter subroutine calculated converter input speed and power as a function of output speed and power. Torque converter performance was defined by plots of torque ratio and capacity factor versus speed ratio (Figure 6-5). The curves are applicable to a standard 11.75-in.-diam unit, having a stall/torque ratio of 2:1.

In addition to the normal operating mode where the transmission remained coupled to the engine at all times, the following optional coupling modes could be specified:

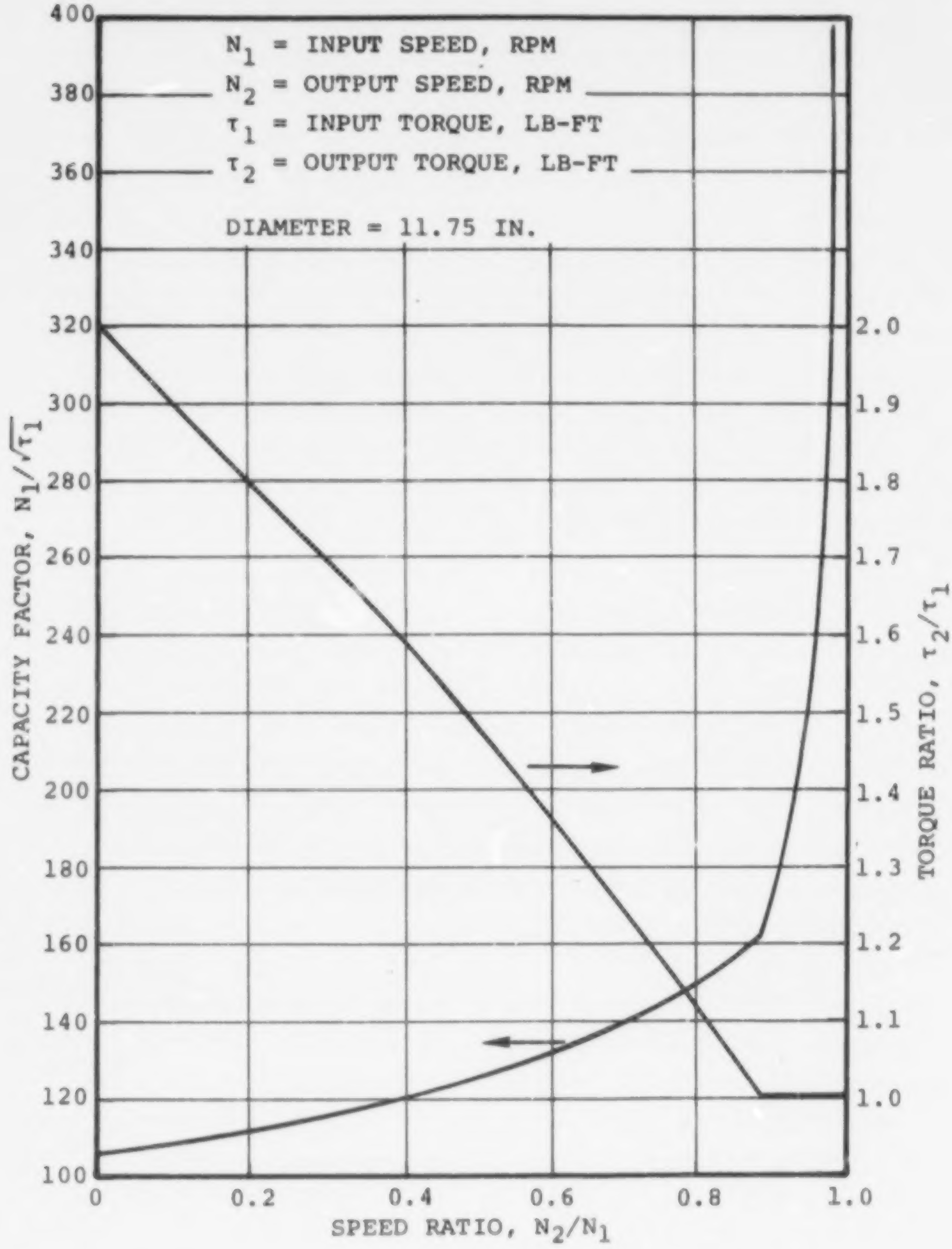
(a) Normal torque converter operation, decoupled when vehicle was stopped

(b) Slipping clutch operation below minimum engine speed, direct coupling at higher speeds

The engine was always confined to a designated speed range, regardless of the coupling mode. If additional "slip" was required to maintain minimum engine speed in the lowest gear, this would override the torque converter characteristics. The slipping-clutch feature was imperative for the single-shaft gas turbine due to engine torque/speed characteristics and limited percent speed range.

6.4 ACCELERATION AND DECELERATION

Acceleration time, distance, and fuel consumption were computed by a finite difference method, using 10 arbitrary vehicle velocity increments per route segment. A sinusoidal speed distribution was assumed to secure greater accuracy at the beginning and end of the acceleration interval. The number of increments represented a compromise between accuracy and computer time. Vehicle acceleration time for each speed increment was computed by determining full throttle engine power at the average engine speed, deducting energy absorbed by



TORQUE CONVERTER CHARACTERISTICS (OAP DATA)

FIGURE 6-5

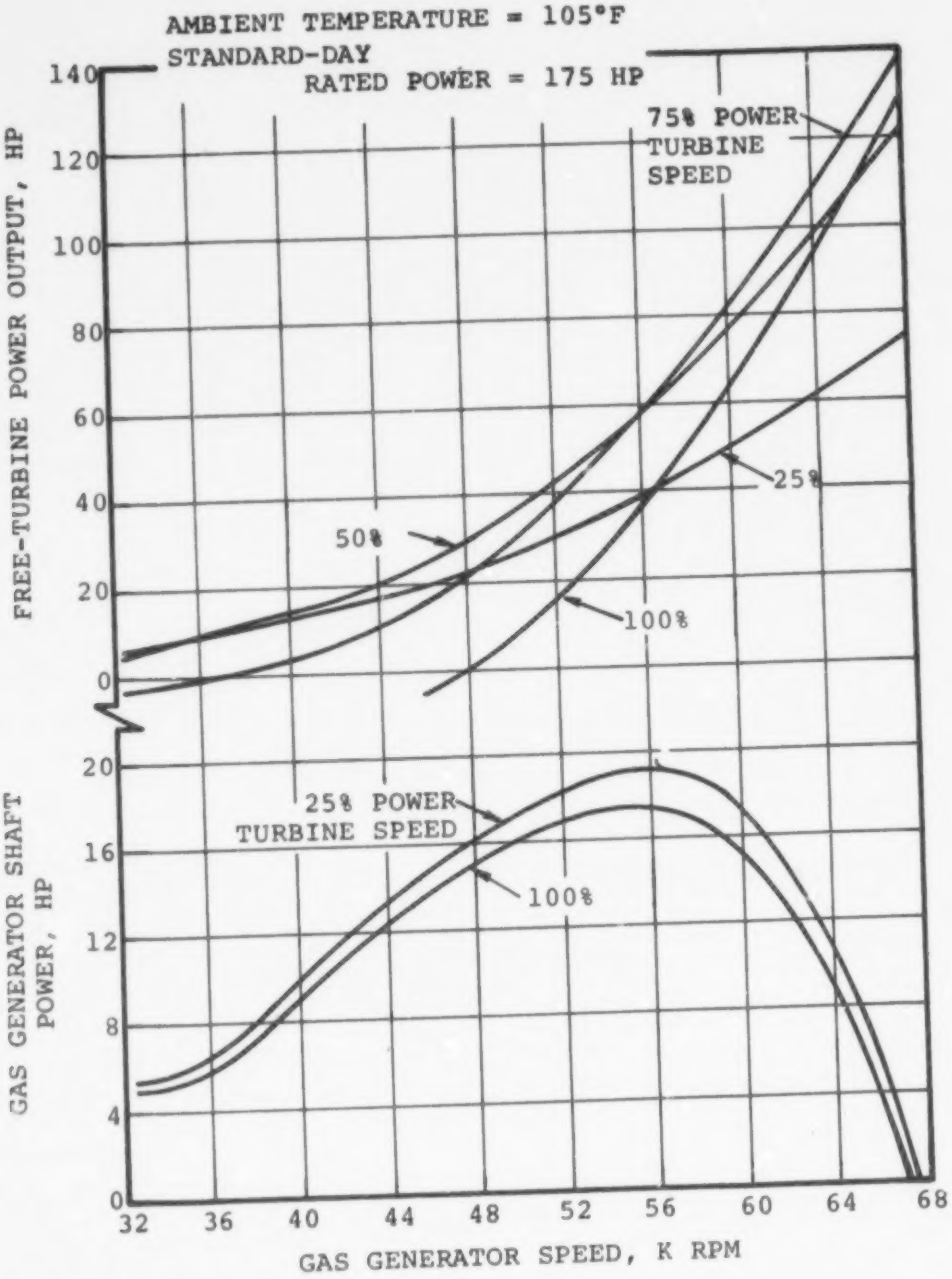
the engine, and applying the remaining excess power to accelerate the vehicle with allowance for transmission losses. Engine rotational energy was deducted only to the extent that it exceeded that previously computed in a given route segment, when an infinitely variable transmission was used.

A minimum engine speed, exceeding idle, to which the engine must be accelerated before initiating vehicle acceleration, could be specified for the single-shaft engine. This permitted a higher initial torque to be transmitted to the wheels, while the clutch was slipping. A time integration of engine speed, fuel consumption, and distance was made at constant vehicle velocity, until the desired engine speed was attained.

The inertial energy required to accelerate the engine (for both the power turbine and gas generator of the free-turbine engine) was taken into account for all vehicle accelerations. The curves plotted in Figure 6-6 illustrate a strong dependence of free-turbine power output on gas-generator speed. The free-turbine acceleration subroutine allowed the gas generator to be accelerated above steady-state speed before free-turbine acceleration, as desired. This feature proved useful for calculating the time required to develop maximum usable engine power output.

On the basis of OAP guidelines specifying the tire adhesion coefficient and weight distribution, maximum acceleration for the performance runs was 1/2 g or 10.96 mph/sec. Since the traction limiting condition would generally not apply except at very low vehicle speeds, the same vehicle acceleration limit could be imposed for each velocity increment with little error.

The computer acceleration subroutine determined the speed ratio yielding maximum transmission output torque at full throttle for each



FREE-TURBINE ENGINE AIII2V MAXIMUM THROTTLE PERFORMANCE

FIGURE 6-6

vehicle speed increment, subject to the engine speed limitations. If the resulting vehicle acceleration exceeded the maximum permissible acceleration, the throttle setting was reduced to scale down the excess power at the wheels in direct proportion to the needed acceleration. This scheme ensured that the engine would accelerate quickly to the speed for producing maximum power, even though the maximum power was not used initially.

Acceleration time, distance, and fuel consumption were integrated for the various speed increments. If the route segment were too short to attain the designated speed limit, a new terminal speed for that requirement was computed and the integration steps repeated. Likewise, if a segment were too short to allow acceleration to the speed limit when a stop was required at the following segment, the intersection speed of the acceleration and braking curves was calculated by a separate subroutine. The computer segment printout displayed the engine conditions prevailing at the maximum speed condition within the route segment.

Vehicle deceleration characteristics were determined by a braking subroutine in which a constant deceleration rate, appropriate for the Federal Register Cycle or tire-adhesion considerations, could be specified. Engine fuel consumption during braking was based on idle speed and accessory power consumption. If a particular segment were too short for slow-down or stop, an appropriate diagnostic message was printed.

6.5 FEDERAL REGISTER DRIVING CYCLE (LA-4 ROUTE)

The AiResearch mission analysis programs required that the route segments be defined by length, grade, and speed limit, or by cumulative distance, elevation, and speed limit. When a segment had a speed limit of zero, the length entry was interpreted as the time interval, in minutes required to traverse the segment. The dynamometer cycle, published in the November 10, 1970, Federal Register, simulated a 7.43-mi route near downtown Los Angeles. The entire route could be closely approximated by a large number of constant-speed-limit segments, with all accelerations and decelerations limited to 3.5 mph/sec. The 143-segment route profile of Table 6-1 is an approximation of the Federal Register Driving Cycle modeled for the AiResearch mission analysis programs.

6.6 UNIFORM SIMPLIFIED DRIVING CYCLE

A uniform simplified driving cycle specified by the OAP simulated vehicle operation for 105,000 mi and 3500 hr according to the following schedule:

Route	Average Speed, mph	Hours	Percent of Total Time
Federal Driving Cycle	19.84	1750	50
Simplified Suburban Route (equal times at constant speeds of 20, 30, and 40 mph)	30.00	1150	33
Simplified Country Route (equal times at constant speeds of 50, 60, and 70 mph)	60.00	600	17

The average mpg obtained on this driving cycle was used to compute the total fuel costs for each of the three candidate cycles used in Section 10, Economic Analysis.

TABLE 6-1

SIMULATED DYNAMOMETER DRIVING SCHEDULE (November 10, 1970, Federal Register)

Segment Number	Length, mi	Speed Limit, mph	Segment Number	Length, mi	Speed Limit, mph
1	0.3330	0.	41	0.0584	36.0
2	0.0215	17.3	42	0.1004	36.1
3	0.0293	22.5	43	0.0373	31.4
4	0.0379	14.9	44	0.0321	17.3
5	0.0520	22.7	45	0.0119	8.8
6	0.0266	15.8	46	0.1000	0.0
7	0.0547	25.0	47	0.0181	19.1
8	0.0972	25.0	48	0.0443	30.0
9	0.0439	26.4	49	0.0489	27.4
10	0.0497	29.8	50	0.0299	14.0
11	0.1353	30.4	51	0.3170	0.0
12	0.0689	31.0	52	0.0257	23.1
13	0.0440	17.5	53	0.0797	35.6
14	0.0122	9.4	54	0.2752	35.4
15	0.6500	0.0	55	0.0532	19.1
16	0.0516	25.8	56	0.0145	9.9
17	0.0565	25.4	57	0.1000	0.0
18	0.0430	17.2	58	0.0344	19.0
19	0.0745	37.3	59	0.0373	24.5
20	0.0974	47.5	60	0.1048	25.1
21	0.1443	47.3	61	0.0324	12.5
22	0.1873	51.9	62	0.2833	0.0
23	0.2011	55.6	63	0.0223	17.0
24	0.1408	56.2	64	0.0852	17.1
25	0.0900	54.0	65	0.0708	21.2
26	0.0896	53.7	66	0.0419	27.0
27	0.1155	51.9	67	0.0322	13.5
28	0.1497	56.0	68	0.4333	0.0
29	0.1053	51.5	69	0.0287	19.6
30	0.0984	50.6	70	0.0394	25.6
31	0.0955	48.1	71	0.0654	26.2
32	0.0739	41.5	72	0.0350	13.1
33	0.0691	31.5	73	0.2333	0.0
34	0.0334	29.0	74	0.0466	19.2
35	0.0365	18.5	75	0.0495	19.8
36	0.0194	9.3	76	0.0311	9.9
37	0.2330	0.0	77	0.0333	0.0
38	0.0209	20.0	78	0.0276	19.6
39	0.0434	30.0	79	0.0644	28.3
40	0.0639	34.6	80	0.0394	28.3

TABLE 6-1 (Contd)

Segment Number	Length, mi	Speed Limit, mph	Segment Number	Length, mi	Speed Limit, mph
81	0.0347	21.5	113	0.0404	11.1
82	0.0235	10.3	114	0.0277	9.1
83	0.0333	0.0	115	0.0055	4.5
84	0.0180	17.5	116	0.0333	0.0
85	0.0423	28.0	117	0.0328	20.0
86	0.1482	28.1	118	0.0359	21.5
87	0.0709	31.9	119	0.1718	25.8
88	0.0929	33.4	120	0.0376	12.9
89	0.1172	30.2	121	0.2667	0.0
90	0.0538	19.2	122	0.0360	23.5
91	0.1018	27.8	123	0.0323	11.8
92	0.0480	28.8	124	0.1667	0.0
93	0.0439	24.5	125	0.0093	12.0
94	0.0661	28.4	126	0.0215	12.9
95	0.0716	28.7	127	0.0294	21.0
96	0.1439	27.3	128	0.1109	21.0
97	0.0503	25.9	129	0.0249	10.5
98	0.0326	21.6	130	0.1333	0.0
99	0.0478	24.4	131	0.0014	1.0
100	0.1589	24.9	132	0.0134	10.5
101	0.0327	12.4	133	0.0096	8.6
102	0.0667	0.0	134	0.0290	21.8
103	0.0600	26.3	135	0.0853	23.6
104	0.0774	27.9	136	0.0698	25.1
105	0.0475	21.5	137	0.0551	27.0
106	0.0626	22.6	138	0.0292	14.2
107	0.0621	24.8	139	0.4167	0.0
108	0.0444	22.9	140	0.0302	19.5
109	0.0211	11.5	141	0.0647	21.2
110	0.5000	0.0	142	0.0302	10.6
111	0.0448	24.0	143	0.0833	0.0
112	0.0893	26.9			

SECTION 7

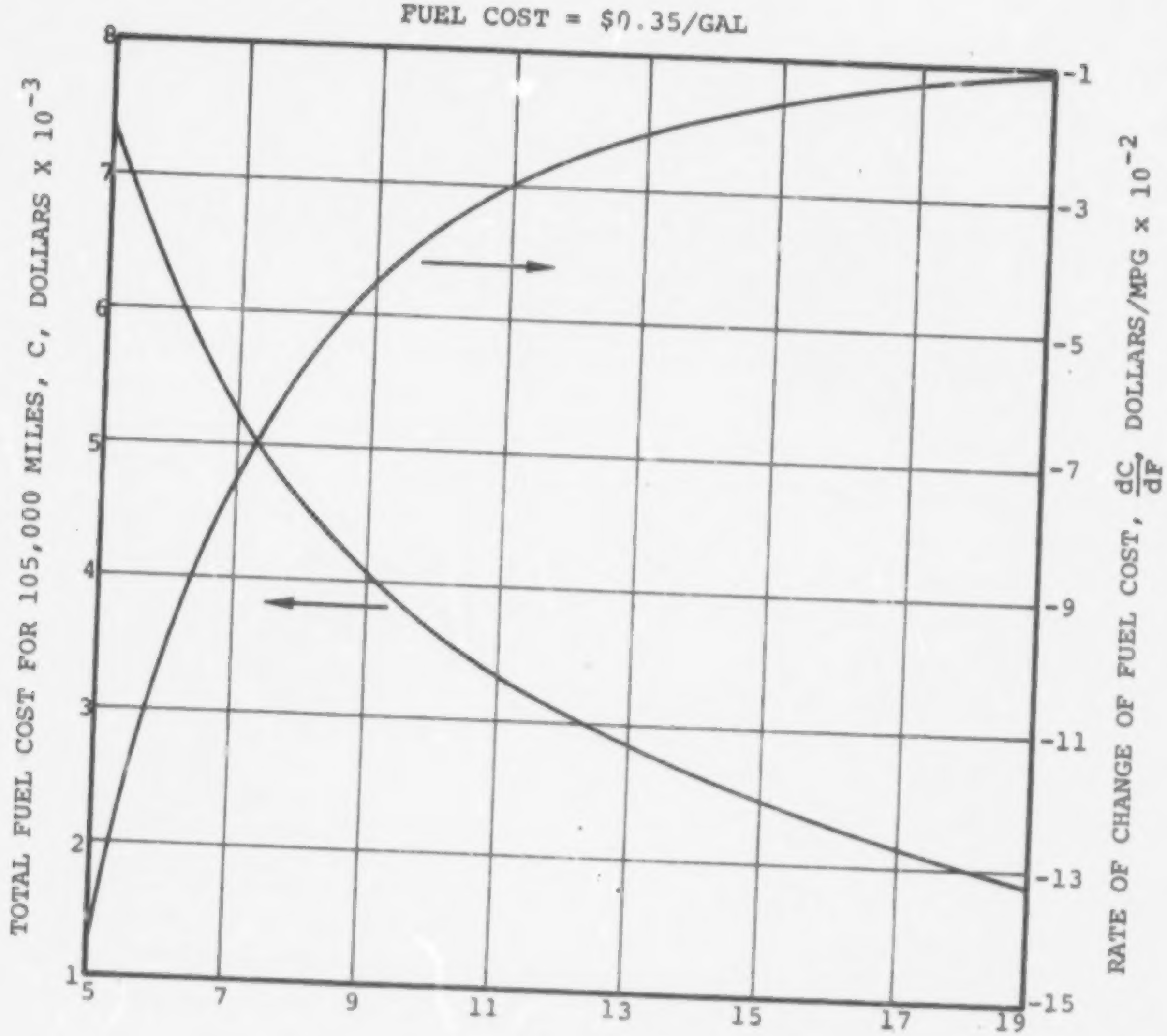
CANDIDATE CYCLE SELECTION

7.1 SELECTION CRITERIA

Selection of three candidate cycles evolved primarily through the process of (1) keeping the configurations simple to minimize initial cost, and (2) improving the average fuel consumption of the two basic engines considered (the single-shaft and free-turbine) by adding combinations of cycle modifiers to the simplest configuration of each. Other factors, such as emissions and initial vehicle cost, were considered only in a qualitative sense, since an accurate means of predicting emissions did not exist and time was not available for making cost estimates, using design layouts. Also considered were factors such as complexity, reliability, maintainability, and weight. Fuel consumption was the only variable for which quantitative values were available. However, fuel consumption is, next to emissions, the most important variable in the optimization study, since cost-of-ownership is greatly affected by small changes in fuel consumption, especially at low values of average miles per gallon (Figure 7-1). For example, if the average fuel consumption is 7 mpg, small increases in this value are worth $750/mpg to the owner in terms of cost-of-ownership, all other variables being equal. This type of consideration motivates a search for improved fuel consumption as long as the added cost and complexity is outweighed by the miles per gallon improvement. Also note that emissions are directly proportional to fuel consumption, all other factors being equal.

7.2 SELECTION OF THREE ENGINES

The engines in the mission analysis procedure are listed in their final form and in almost the chronological order in which they were studied (Table 7-1). In general, as degrees of complexity were added,



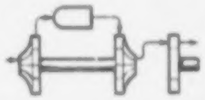
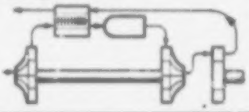
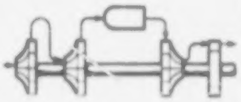
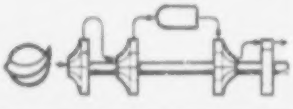
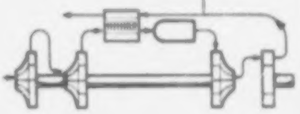
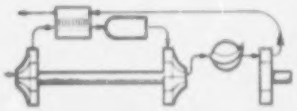
AVERAGE FUEL CONSUMPTION, F , MPG

TOTAL FUEL COSTS AS A FUNCTION OF FUEL CONSUMPTION

FIGURE 7-1

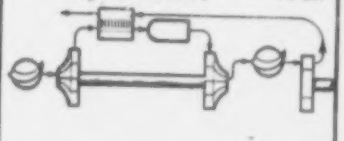
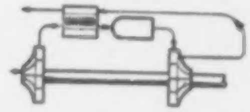
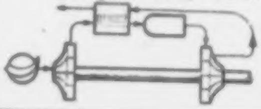
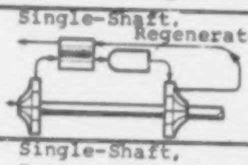
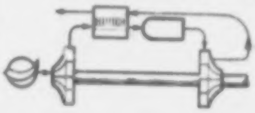
TABLE 7-1

ENGINES ANALYZED

Engine Configuration	Maximum Cycle Pressure Ratio	Design-Point Regeneration Effectiveness, ϵ_R	Design Point shp, 60°F	FDC mpg, 85°F	USEDC mpg, 85°F	Comment
1. Free-Turbine 	8.0	--	150+	5.0	--	
2. Free-Turbine, Recuperated 	8.0	0.85	150+	8.4	12.2	
3. Single-Shaft 	12.0	--	160	6.3	8.0	
4. Single-Shaft, VIGV* 	12.0	--	160	6.3	8.0	IGVs do not improve unregenerated cycle. Idle W_F decreased by 6 percent.
5. Free-Turbine, Recuperated, Bypass 	10.0	0.70 at 60 percent power	150+	7.9	--	MPG based on min sfc vs shp curve compared with Cycle 2.
6. Free-Turbine, Recuperated, VPTN** 	6.7	0.75	175+	10.0	12.7	
		0.85	175+	13.6	17.1	

*Variable inlet guide vanes
**Variable, power turbine nozzles

TABLE 7-1 (Contd)

Engine Configuration	Maximum Cycle Pressure Ratio	Design-Point Regeneration Effectiveness, ϵ_R	Design Point shp, 60°F	FDC mpg, 85°F	USEDC mpg, 85°F	Comment
7. Free-Turbine, Regenerated, VIGV, VPTN 	4.6	0.90	175	16.4	19.9	No power boost.
			135	18.1	21.6	Power boost for maximum acceleration.
8. Single-Shaft, Recuperated 	6.4	0.80	165	6.0	9.5	
		0.80	120	8.1	12.6	Power boost for maximum acceleration.
9. Single-Shaft, Recuperated, VIGV 	6.4	0.85	125	14.3	20.0	Power boost for maximum acceleration.
10. Single-Shaft, Regenerated 	4.6	0.90	108	9.6	14.9	Power boost for maximum acceleration.
11. Single-Shaft, Regenerated, VIGV 	4.6	0.90	108	17.0	22.1	Power boost for maximum acceleration.
			155	14.0	19.7	
12. V-8 Spark Ignition	-	-	175	12.5	14.8	

fuel consumption continued to decrease, or conversely, the average miles per gallon increased. The design-point shaft horsepower for each of the engines is the horsepower that is required to just meet all acceleration performance requirements. The difference in shaft horsepower values required for the various engines indicates the effects of engine inertia and design speed on engine horsepower and, as a result, on fuel consumption.

The first engine studied was the simple free-turbine cycle without variable geometry or regeneration. With this engine, the vehicle fuel economy on the Federal Driving Cycle was only 5 mpg. The next step was to add a cycle modifier--a recuperator--to improve the off-design performance of the cycle. The fuel economy was considerably improved; however, more miles per gallon appear desirable (Figure 7-1) to reduce cost-of-ownership.

A high-pressure ratio, single-shaft engine with no cycle modifiers was next analyzed. To obtain a somewhat reasonable mechanical design for this cycle, two stages of compression and a two-stage turbine were chosen to obtain high compressor and turbine efficiencies (relative to single-stage values) and to reduce the turbine disc and blade stresses by reducing the physical speed (again compared to single-stage). The engine acceleration time of 1.5 sec could not be met, however, vehicle performance requirements were achieved. The fuel economy was poor compared to that for the free-turbine recuperated engine and compared to the 1970 spark-ignition engine. Because of the high fuel consumption cost, the average consumer could not afford to drive such a vehicle.

A bypass recuperator cycle was analyzed as a candidate for reducing the size of the recuperator and, thus, effecting a reduction in the manufacturing and consumer cost. The bypass-power point was selected as 65 hp for this engine. The recuperator for the non-bypass engine was to have an effectiveness of 0.85 at the design-point. The recuperator for the bypass engine was designed with the same effectiveness at the 65-hp point as that for the non-bypass recuperator. The effect of varying the core pressure loss at this power point was then determined. By increasing the core pressure loss to 20 percent, a weight saving of 29 percent could be achieved. However, the cost saving for such a design is not expected to be significant.

For example, the non-bypass recuperator for the above engine weighs 68 lb for a metallic core. At 20-percent core-pressure drop for the 65-hp point, the corresponding weight is 48 lb for the bypass system, or a saving of 20 lb. Assuming that the manufacturing cost for the recuperator is $1.50/lb, the recuperator cost saving would be $30. However, the costs of a high-temperature valve, additional ducting, and possibly additional control system have to be subtracted for the $30.00 saving. Also, the bypass engine will have a higher sfc at all power conditions below the bypass point, since the cycle pressure loss is higher at each comparable operating condition. Thus, the fuel economy of the bypass engine will be reduced. Because of the additional complexity, the bypass engine would be less reliable, causing an increase in maintenance costs. Therefore, the bypass system appears to offer little potential as a significant cost reduction scheme.

The three engines with the greatest potential for the automobile are the regenerated free-turbine, the regenerated single-shaft, and the recuperated single-shaft, Cycles 7, 9, and 11, respectively (Table 7-1).

For each of the candidate engines, the heat exchanger effectiveness value was chosen as high as could be considered practical, with respect to volume and greatest dimension. The limiting values of effectiveness were chosen as 0.85 for recuperators and 0.90 for regenerators.

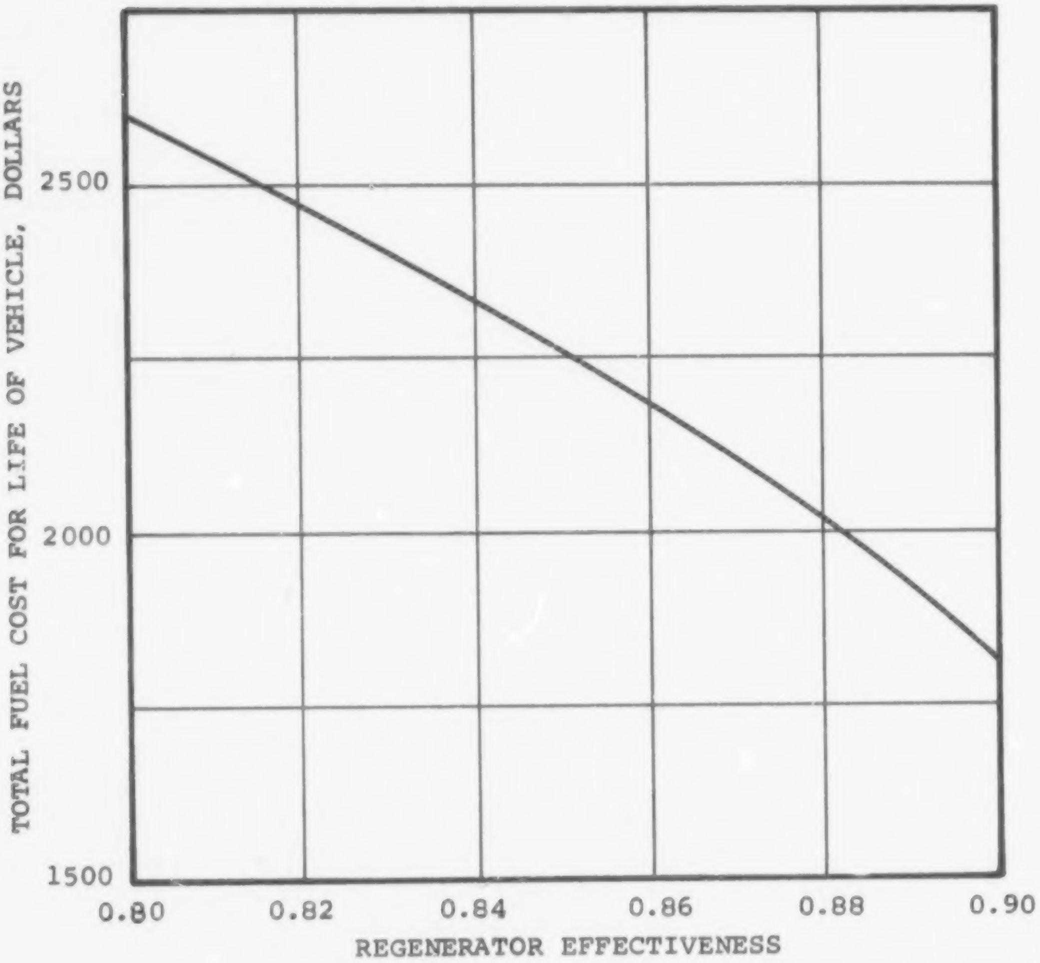
The manufacturing costs of a heat exchanger and related parts vary as a function of heat-exchanger effectiveness, but the effect on vehicle cost-of-ownership is smaller than the change in fuel cost for the life of the vehicle for changes in effectiveness in the range of 0.80 to 0.90. This is illustrated in Figure 7-2 for the single-shaft regenerated engine. For example, reducing the effectiveness from 0.90 to 0.85 increases the life-time fuel costs by $425, whereas the total cost of the regenerator system for this engine is only $57, including regenerator core, seals, bearings, drive gears, and drive motor (Table 8-3). A similar relationship exists for recuperated engines, there is no crossover point between cost increase in fuel and cost savings in recuperator in the effectiveness range near 0.85.

7.3 PERFORMANCE OF CANDIDATE ENGINES

Performance requirements for the vehicle and engine are specified in the OAP document "Vehicle Design Goals - Six-Passenger Automobile," included as Appendix 1 of this report. These requirements are shown on Table 7-2.

The requirement of accelerating the engine from idle speed to a speed that provides maximum output torque (100-percent engine speed for a single-shaft engine and 100-percent gas generator speed for a free-turbine configuration) is incompatible with the vehicle acceleration requirements. For a given engine configuration, this time decreases as the engine is scaled down to a lower maximum output horsepower, whereas the vehicle acceleration time increases for the various maneuvers. Thus, an engine can be designed that will accelerate the vehicle at the traction limit with a response time to obtain the required output power of less than 1.0 sec, but with a time to accelerate from idle to maximum torque of greater than 1.5 sec. An analysis of this contradiction is given in Appendix 3. A change is recommended in this engine acceleration requirement to specify a vehicle acceleration rate that must be obtained in a given time, such as

NOTES:
1. SINGLE-SHAFT REGENERATED ENGINE (108 HP RATED)
2. TRACTION TRANSMISSION
3. FUEL COST = $0.36/GAL
4. TOTAL MILAGE FOR LIFE OF VEHICLE = 105,220
5. AMBIENT TEMPERATURE = 85°F



EFFECT OF REGENERATOR EFFECTIVENESS ON
COST OF FUEL FOR LIFE OF VEHICLE

FIGURE 7-2

TABLE 7-2

PERFORMANCE REQUIREMENTS*

Ambient Temperature, °F	Standing Start		Merging Traffic 25-70 mph, sec	DOT High Speed Pass, sec	30% Grade Velocity, mph	5% Grade Velocity, mph	Idle to Maximum Torque, sec	Idle Fuel Flow, Percent of Maximum
	0-440 ft, sec	0-60 mph, sec						
85	10.0	13.5	15.0	15.0	15.0	70.0	1.5	14
-20 to 105 (excluding 85)	10.5	14.3	15.8	15.8	14.2	66.5	1.6	14

*These requirements are as specified in "Vehicle Design Goals - Six Passenger Automobile" (Appendix 1).

3 mph/sec in 0.5 sec. (This is an example only and is not necessarily a recommended value.) This maneuver might be referred to as the "standing-start initial acceleration rate" for descriptive purposes.

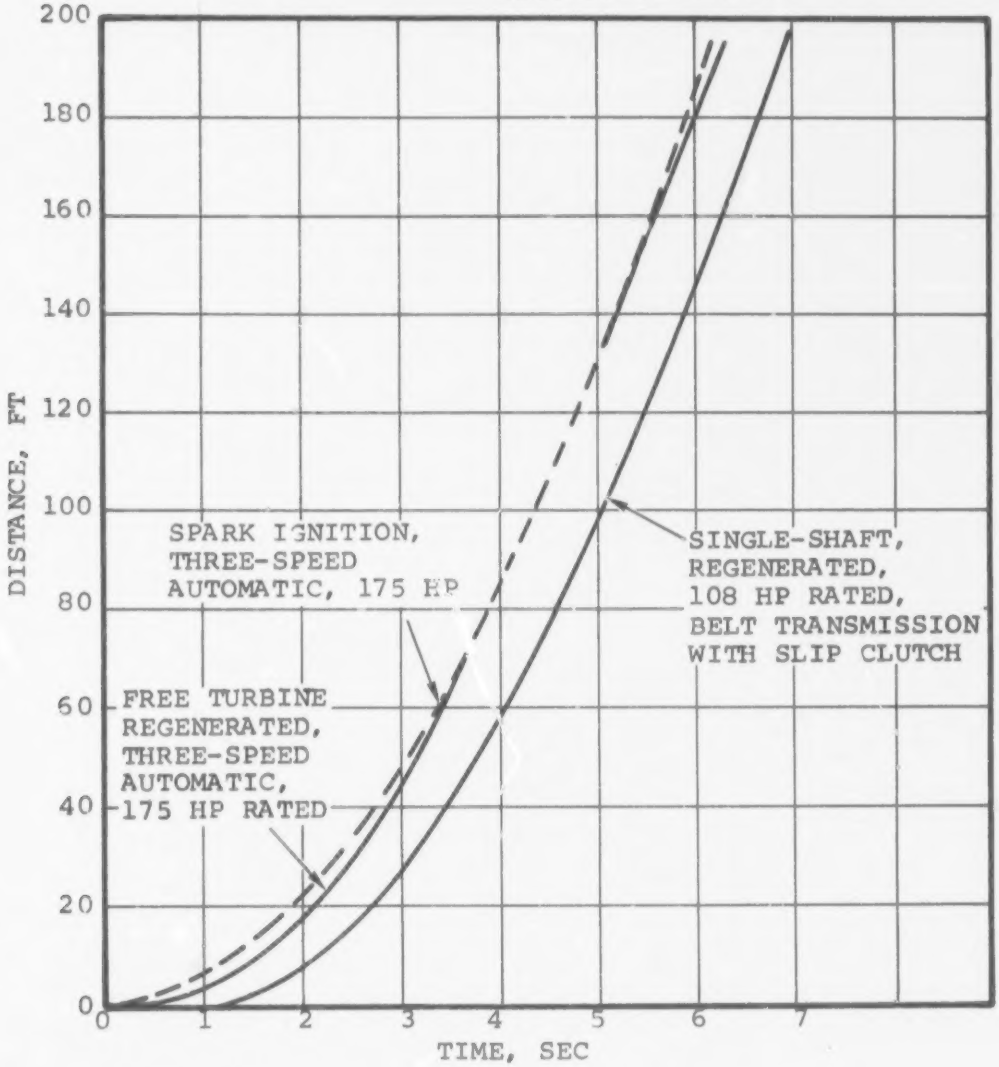
The gas turbine automobile should respond to throttle position in a manner similar to the spark-ignition (SI) automobile, to obtain the greatest consumer acceptance and to minimize safety problems associated with the vehicle.

Previous works have shown that the standing-start acceleration response is much slower with the gas turbine than with the spark-ignition engine. However, the difference in standing-start performance can be essentially eliminated by proper design of the system. Figure 7-3 compares the standing-start performance of two candidate gas turbines and a spark-ignition engine. The free-turbine performance is very close to the SI performance and is achieved by idling the gas generator at 73-percent speed. Fuel economy is achieved with this engine by using variable inlet guide vanes. The standing-start performance of the single-shaft engine with slipping-clutch lags both the SI and free-turbine engines by almost 1 sec. The single-shaft performance can be improved by replacing the slipping-clutch with a torque converter (Figure 7-4), in which the single-shaft engine is essentially the same as the SI engine.

Performance of the single-shaft regenerated engine is shown on Table 7-3 for ambient temperatures between 30° and 105°F for the 108-hp rated engine and for a 105°F day for the 155-hp rated engine. For this engine, 94-percent speed is reached in 1.4 sec, which provides an initial vehicle acceleration rate of 10 mph/sec. As the acceleration time increases from 0.85 to 1 sec, the speed increases from idling at 60 percent to 96 percent, depending on temperature variance from 59° to 105°F.

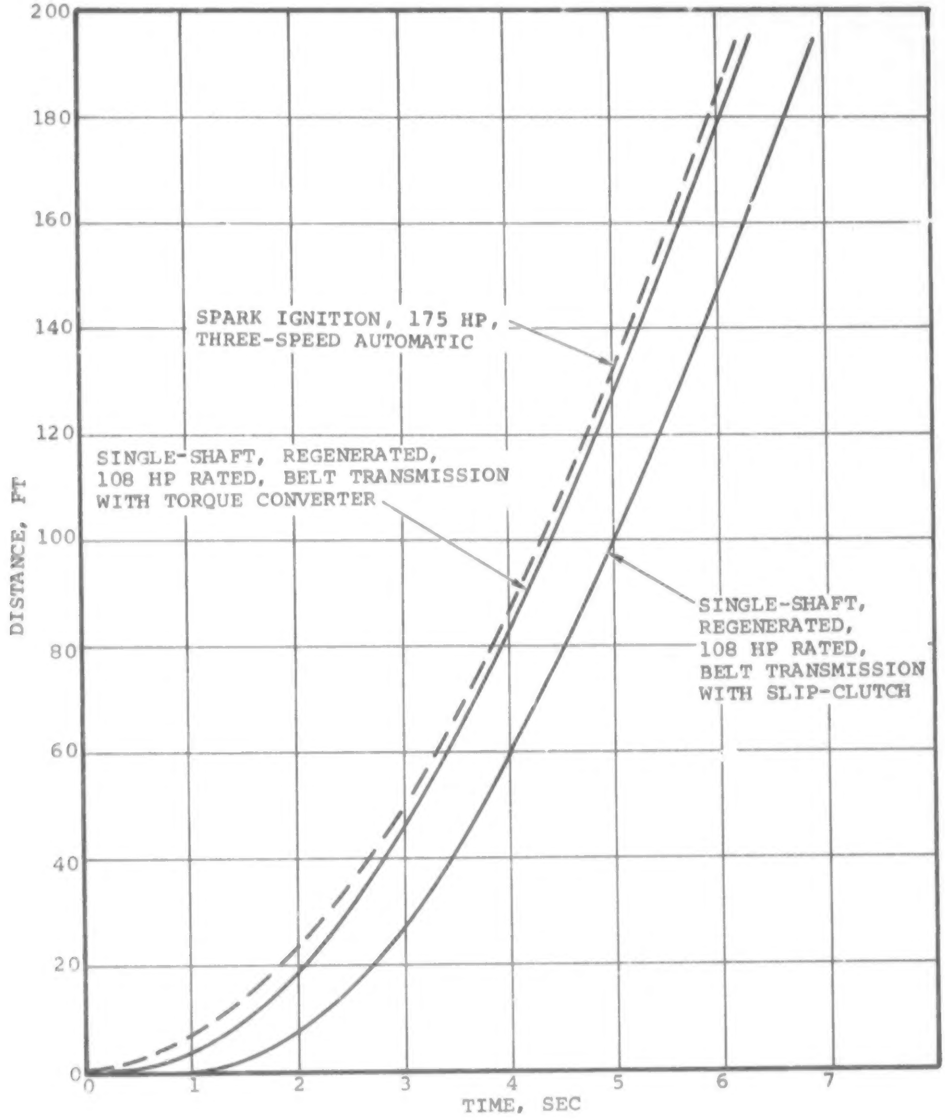
NOTES:

1. $T_{AMB} = 105°F$
2. VEHICLE WEIGHT = 4000-LB TURBINE ENGINE
   = 4300-LB SI ENGINE



STANDING-START ACCELERATION PERFORMANCE

FIGURE 7-3



NOTES:
1. T_{AMB} = 105°F
2. VEHICLE WEIGHT = 4000-LB TURBINE ENGINE
= 4300-LB SI ENGINE

STANDING-START ACCELERATION PERFORMANCE

FIGURE 7-4

TABLE 7-3

PERFORMANCE OF SINGLE-SHAFT REGENERATED ENGINE (4000-LB VEHICLE, 4-HP, ACCESSORY LOAD FOR ACCELERATION)

Ambient Temperature °F	Power Boost, %	Design Speed (includes Power Boost), max shp	Acceleration Time					Grade Velocity, mph		Idle Fuel Flow,** Percent of Maximum
			Idle to Maximum Torque Speed, sec	0-440 ft, sec	0-60 mph, sec	25-70 mph, sec	DOT Pass, sec	5%	30%*	
105	30	110 (1)	1.0	10.4	13.4	15.3	15.1	90	25+	5.6
85	24	120 (1)	0.9	9.9	11.3	12.5	14.5	↓	↓	5.6
59	9	120 (1)	0.9	9.9	11.2	12.4	13.5	↓	↓	5.9
30	0	138 (1)	0.85	9.5	10.3	11.5	12.3	↓	↓	6.0
105	0	122 (2)	1.4	10.5	12.0	13.2	14.8	↓	↓	5.3
105	30	123 (3)	1.3	10.4	12.6	14.3	15.0	↓	↓	5.8

*4700-lb vehicle
**Obtained with 1.3-hp accessory load

(1) Engine rated power = 108 shp (sea level, 59°F)
(2) Engine rated power = 155 shp
(3) Engine rated power = 122 shp, vehicle weight = 4600 lb

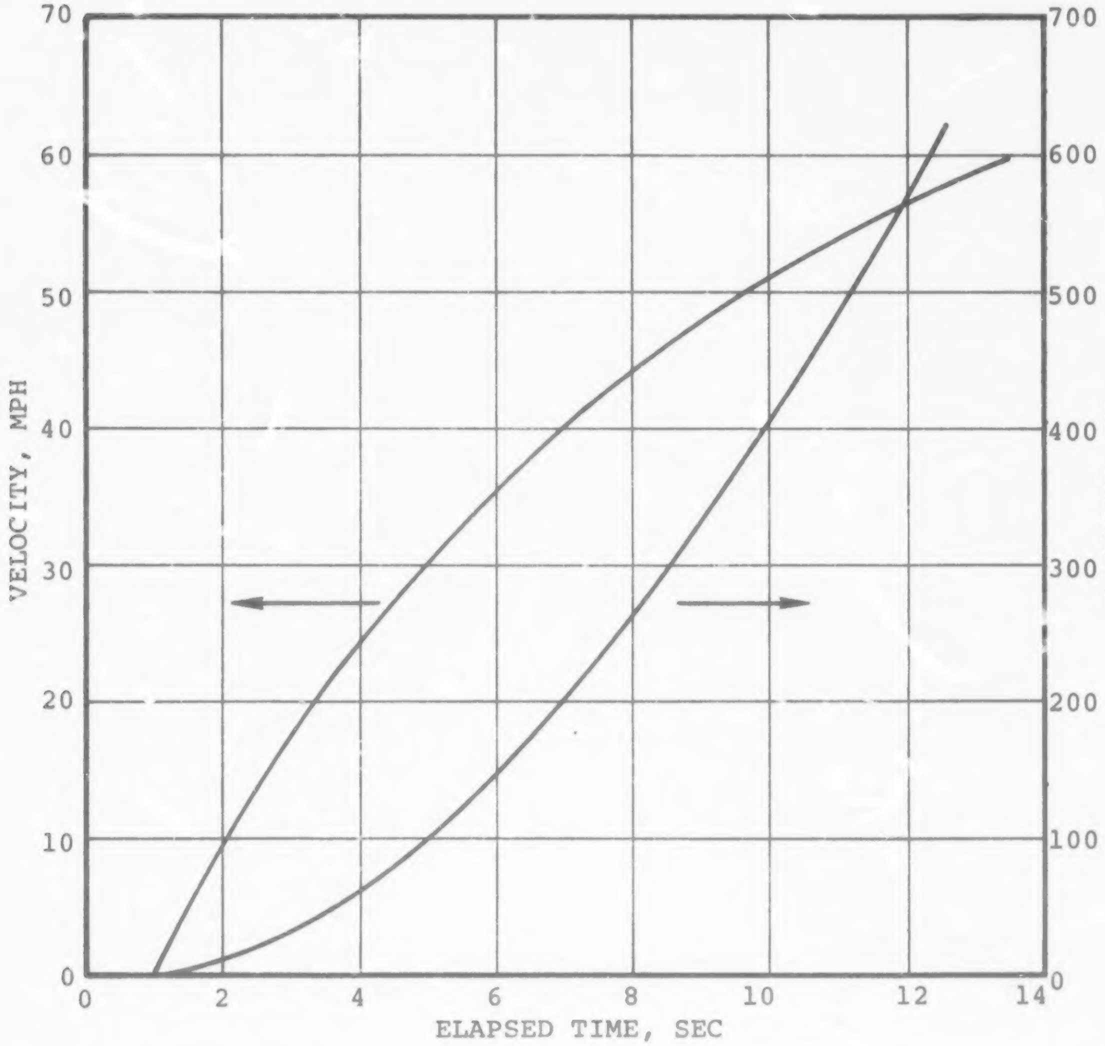
The power boost required for ambient temperatures less than 105°F is minimal, the water flow rate being greatest for the 105°F day. The required water injection will not be more than 1 to 2 percent at the most, for a 59°F day, i.e., the boost obtained with approximately 6-percent injection provides more output power than is needed. Note also that the water injection scheme does not require that an anti-freezing agent be added to the water, since water injection is not required at ambient temperatures near freezing. All that is necessary is a water storage tank that has sufficient volume of elasticity to permit expansion when water freezes.

Also, shown on Table 7-3 is the engine sizing and performance results for an assumed 4600-lb traction transmission vehicle. To meet all performance requirements for this heavier vehicle, the engine power rating must be increased from 108 to 122 shp rated for a 59°F sea-level day, plus 30-percent power boost for 105°F day.

Standing-start acceleration performance for the single-shaft regenerated engine is shown in Figure 7-5, and results for the merging traffic and DOT-pass maneuvers are plotted in Figure 7-6, both for a 105°F day.

Performance of the single-shaft recuperated engine, rated at 125 hp, is shown on Table 7-4. This engine requires a power rating greater than that for the regenerated unit, since the optimum pressure ratio is higher for the lower heat-exchanger effectiveness. Higher pressure ratio means larger diameter compressor and turbine wheels, and the fact that aluminum cannot be used for a high-pressure ratio impeller material because of the resulting high compressor-discharge temperature. These two factors result in a higher inertia engine and, therefore, greater power is required to accelerate.

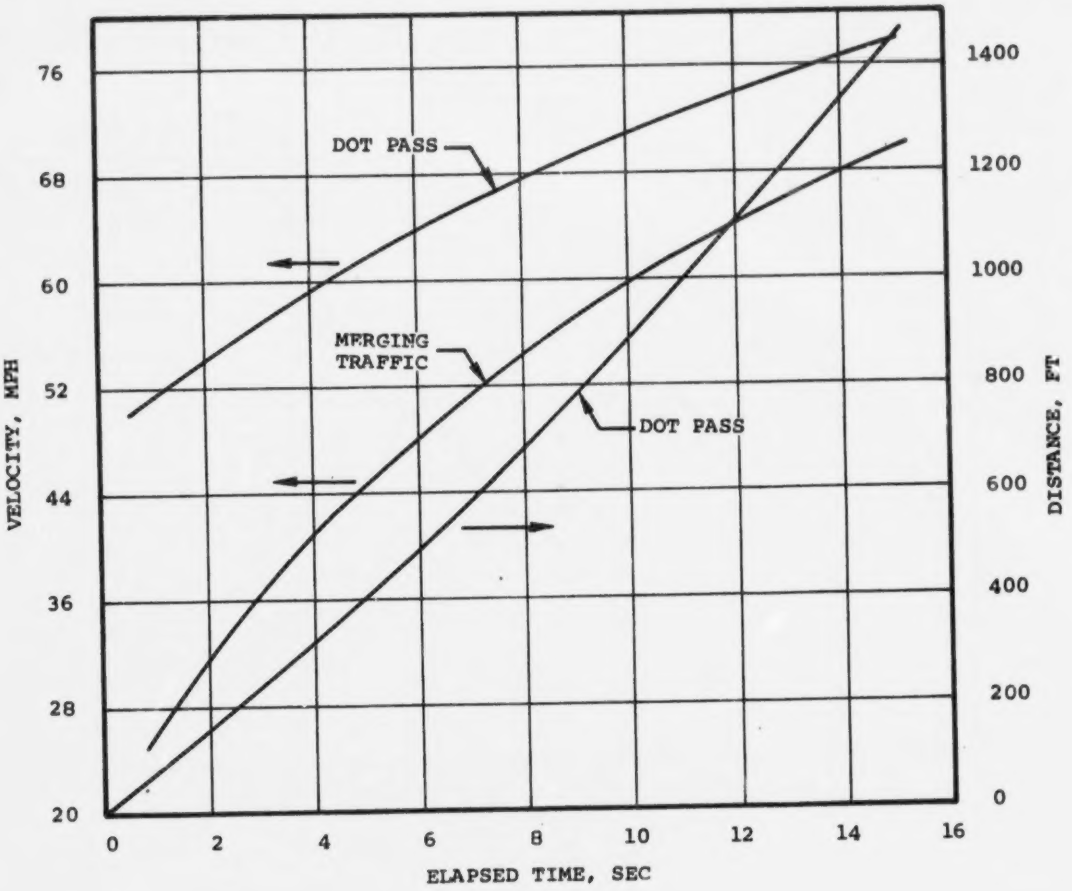
NOTES:
1. AMBIENT TEMPERATURE = 105°F
2. ENGINE RATED AT 108 HP (59°F DAY)
3. ACCESSORY LOAD = 4 HP
4. 4000-LB VEHICLE



STANDING-START VEHICLE ACCELERATION PERFORMANCE
WITH SINGLE-SHAFT REGENERATED ENGINE

FIGURE 7-5

NOTES:
1. AMBIENT TEMPERATURE = 105°F
2. ENGINE RATED AT 108 HP (59°F DAY)
3. ACCESSORY LOAD = 4 HP
4. 4000-LB VEHICLE



MERGING TRAFFIC AND DOT HIGH-SPEED PASS PERFORMANCE WITH SINGLE-SHAFT REGENERATED ENGINE

FIGURE 7-6

TABLE 7-4

PERFORMANCE OF SINGLE-SHAFT RECUPERATED ENGINE, 125 HP RATED
(4000-LB VEHICLE, 4-HP ACCESSORY LOAD FOR ACCELERATION)

Ambient Temperature, °F	Power Boost, %	Design Speed, max shp	Acceleration Time					Grade Velocity, mph		Idle Fuel Flow,** Percent of Maximum
			Idle to Maximum Torque Speed, sec	0-440 ft, sec	0-60 mph, sec	25-70 mph, sec	DOT Pass, sec	30%	5%	
105	30	127	1.6	10.5	12.2	13.6	14.9	25+	90	7.6
85	20	131	1.5	10.4	11.9	13.3	14.2	↓	↓	7.9
59	10	137	1.5	10.3	11.7	12.7	13.7	↓	↓	8.7
30	0	143	1.4	10.0	11.0	12.2	12.8	↓	↓	8.0

*4700-lb vehicle
**Obtained with 1.3 hp accessory load

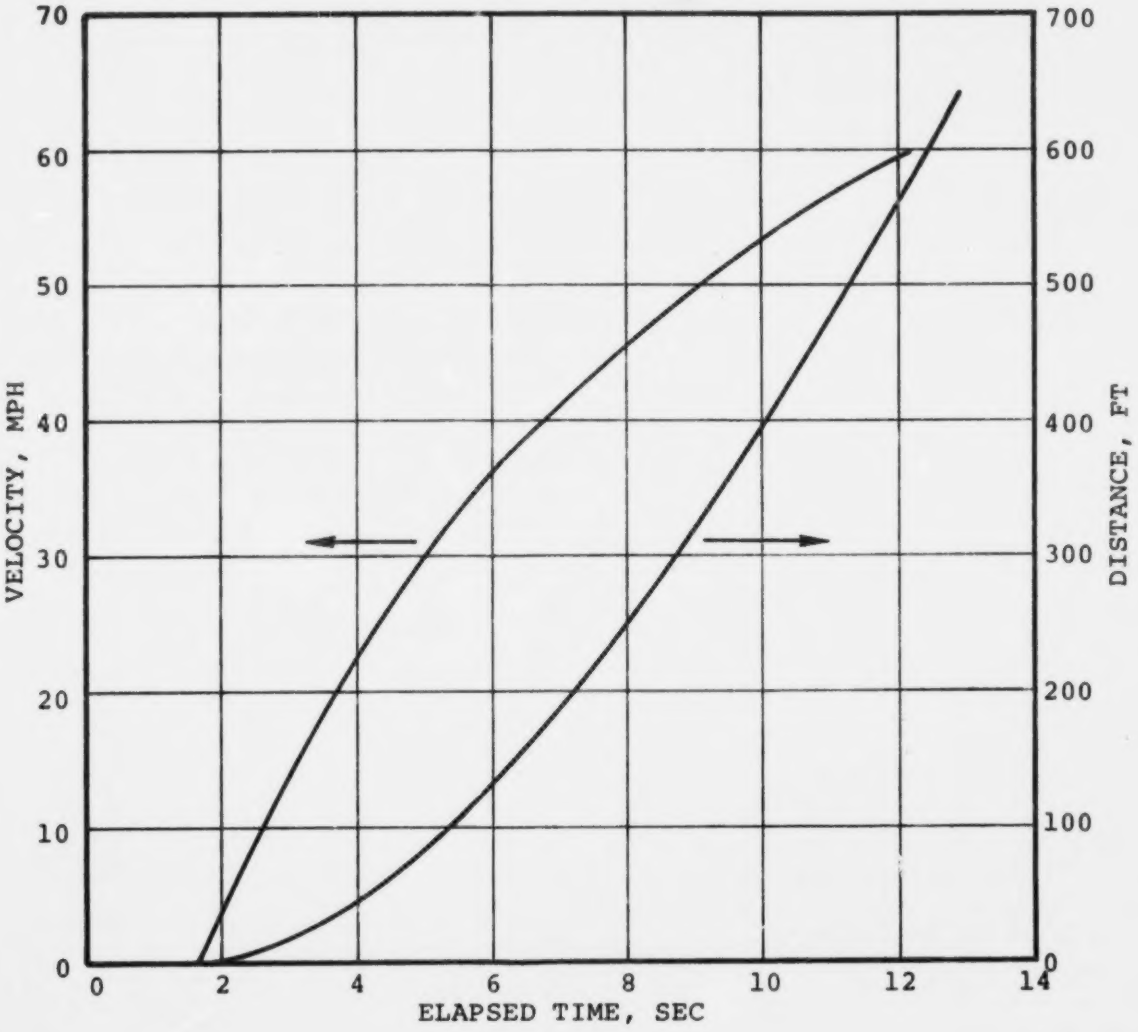
Excessive time to accelerate caused this engine to fail in meeting the idle-to-maximum-torque acceleration requirement. However, this is of little consequence since output power at about 90-percent speed is sufficient to accelerate the 4000-lb automobile at the traction limit.

The elapsed time to reach 440 ft from a standing start slightly exceeds the 10-sec requirement for ambient temperatures of 59° and 85°F. However, if the power boost available for 6-percent water injection is a conservative estimate (Section 5), then possibly the 10-sec requirement would be achieved. The standing-start acceleration, merging traffic, and DOT pass performance results are shown in Figures 7-7 and 7-8, respectively.

Table 7-5 shows performance results for the regenerated free-turbine engine. These results were computed for 105° and 85°F days for the non-power-boosted engine, and for 85°F only for the power-boosted engine. It was found late in the study that the acceleration subroutine in the mission analysis computer program did not properly account for the gas generator mass inertia in computing vehicle and engine acceleration. The computer program was then modified and checked for proper operation by comparing results with hand calculations. Re-analysis of the acceleration performance of the free-turbine engine showed that the cycle had to be modified to meet vehicle requirements. This was accomplished by increasing the maximum output power from approximately 140 to 175 hp and by increasing the idle speed of the gas generator from 49- to 73-percent speed by using variable inlet guide vanes. The latter change necessitated a recomputation of off-design performance of this modified engine. As a result of this late change, time did not permit recalculation of off-design performance for 59° and 30°F. However, it is certain that both the non-power-boosted and the power-boosted engines would continue to meet the acceleration requirements and, therefore, this deletion has absolutely no effect on the study.

NOTES:

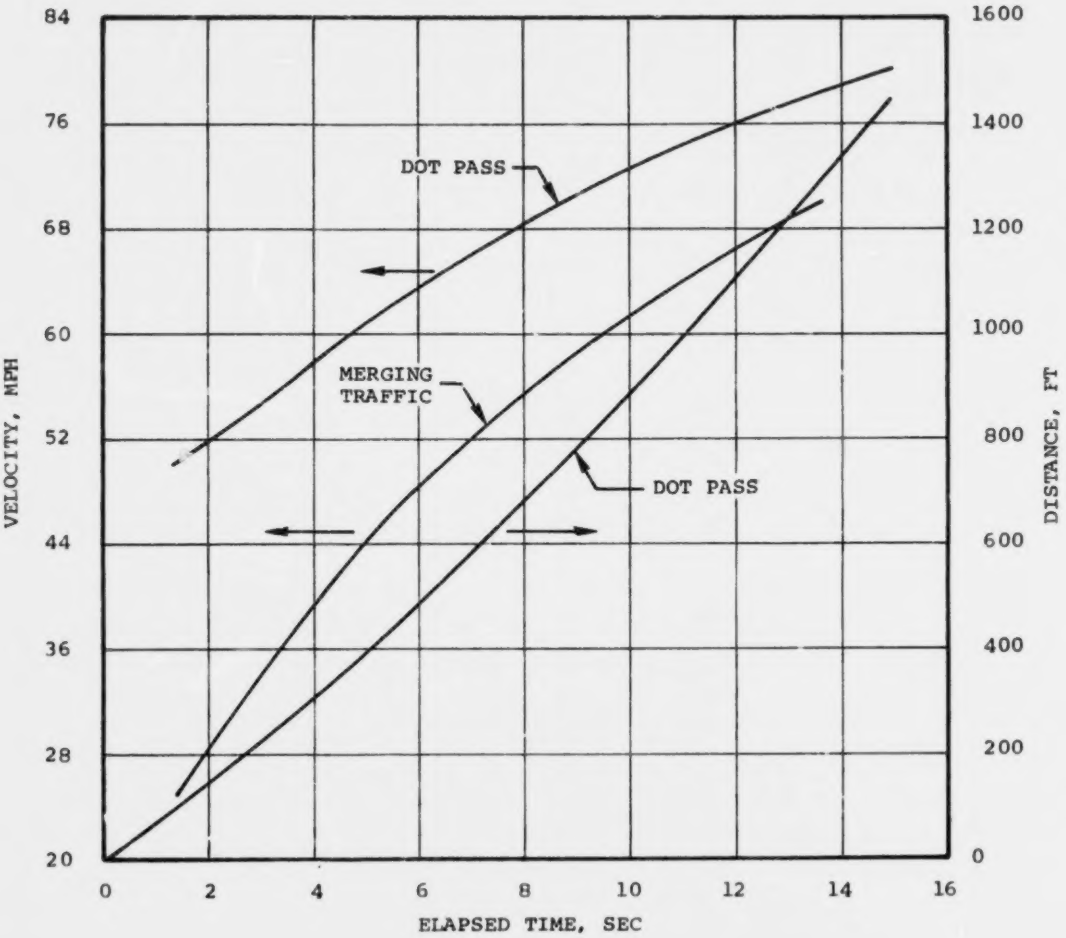
1. AMBIENT TEMPERATURE = 105°F
2. ENGINE RATED AT 125 HP (59°F DAY)
3. ACCESSORY LOAD = 4 HP
4. 4000-LB VEHICLE



STANDING-START VEHICLE ACCELERATION PERFORMANCE
WITH SINGLE-SHAFT RECUPERATED ENGINE

FIGURE 7-7

NOTES:
1. AMBIENT TEMPERATURE = 105°F
2. ENGINE RATED AT 125 HP (59°F DAY)
3. ACCESSORY LOAD = 4 HP
4. 4000-LB VEHICLE



MERGING TRAFFIC AND DOT HIGH-SPEED PASS PERFORMANCE WITH SINGLE-SHAFT RECUPERATED ENGINE

FIGURE 7-8

TABLE 7-5

PERFORMANCE OF FREE-TURBINE REGENERATED ENGINE (4000-LB VEHICLE, 4-HP ACCESSORY LOAD FOR ACCELERATION)

Ambient Temperature, °F	Power Boost, %	Design Speed max shp	Acceleration Time					Grade Velocity, mph		Idle Fuel Flow,** Percent of Maximum
			Idle to Maximum Torque Speed, sec	0-440 ft, sec	0-60 mph, sec	25-70 mph, sec	DOT Pass, sec	30%*	5%	
105	0	136$^{(1)}$	3.0	9.6	11.9	14.6	15.5	25	90	-
85	0	150$^{(1)}$	-	7.2	10.6	12.4	14.7	25	90	5.7
105	30	137$^{(2)}$	-	9.4	11.7	14.4	15.7	25	90	-

*4700-lb vehicle
**Obtained with 1.3-hp accessory load

(1) Engine rated power = 175 shp (sea level, 59°F)
(2) Engine rated power = 135 shp (sea level, 59°F)

The standing-start acceleration, merging traffic, and DOT pass performance results for the free-turbine engine are shown in Figures 7-9 and 7-10, respectively.

7.4 FUEL ECONOMY OF CANDIDATE ENGINES

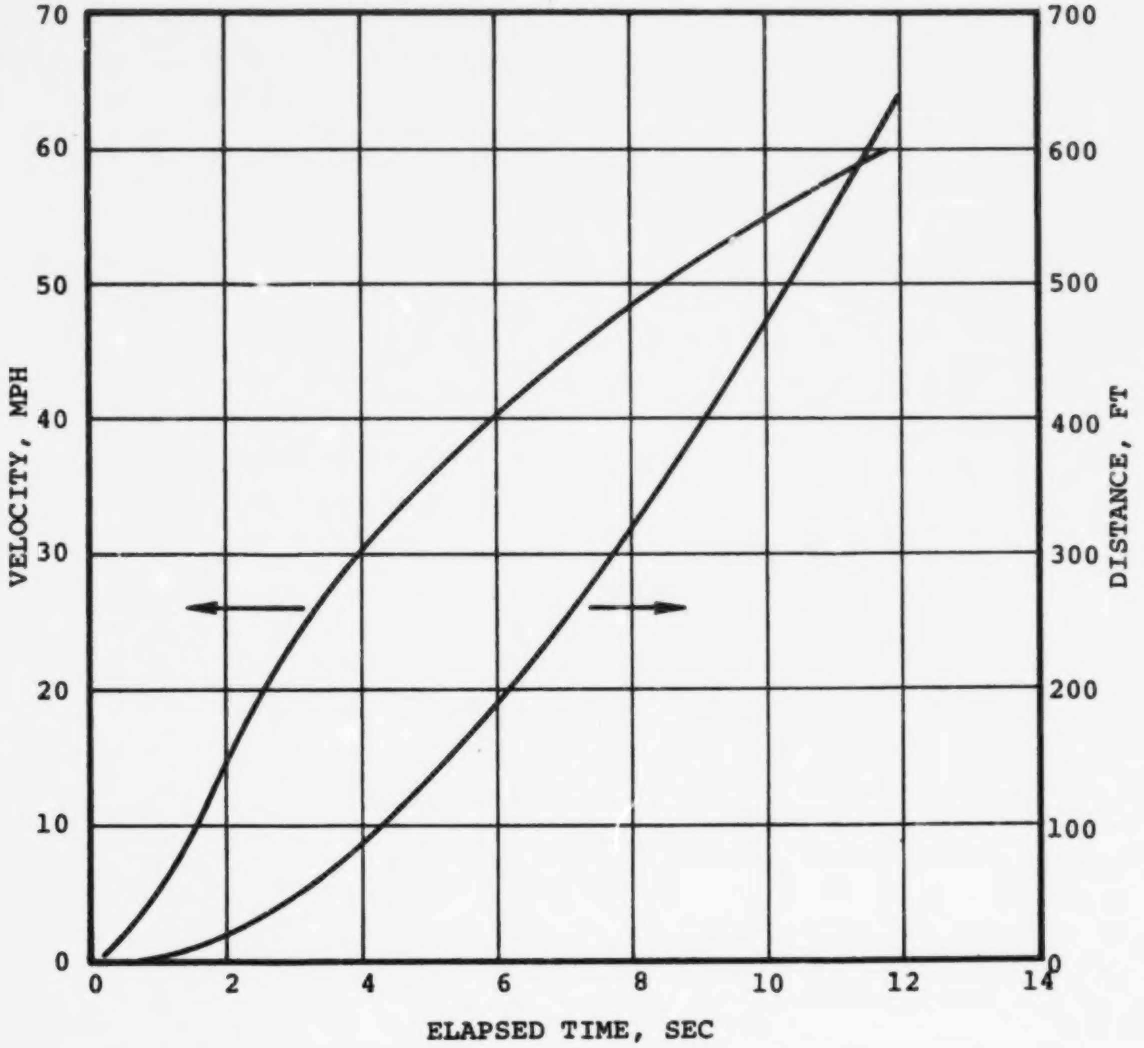
Fuel consumption calculations for all gas turbine engines were based on a fuel having a density of 6.25 lb/gal and a lower heating value of 18,500 Btu/lb (or 11.59×10^4 Btu/gal). The results are conservative in that, if a fuel with a higher heat release, such as JP-5 (12.5×10^4 Btu/gal) were used, the miles-per-gallon values would be increased by approximately 8 percent.

Fuel economy results for the three candidate cycles are shown on Tables 7-6 through 7-8. Results for the spark-ignition engine, using gasoline with a density of 5.87 lb/gal and a lower heating value of 18,700 Btu/lb, are shown on each table for comparison. Because of the late modification to the free-turbine engine (Table 7-8) discussed in Paragraph 7.3, fuel economy was computed for an 85°F day only. This was no serious deletion since the fuel consumption for other temperatures will vary slightly, as indicated on Tables 7-6 and 7-7, and in approximately the same manner as for the single-shaft engines.

Fuel economy is shown (Table 7-6) for a 4600-lb vehicle with single-shaft regenerated engine, appropriately sized to meet acceleration requirements, with a traction transmission. The result of the 600-lb increase in weight is a loss in fuel economy of about 1 mpg on the FDC and a loss of 1.5 mpg for the composite route. Fuel economy of the single-shaft regenerated engine is improved, by using power boost, by about 3 mpg on the Federal Driving Cycle (FDC) and by 2.4 mpg for the composite route. For the free-turbine engine, power boost improves the fuel economy of both the FDC and the composite route by 1.7 mpg (Table 7-8).

NOTES:

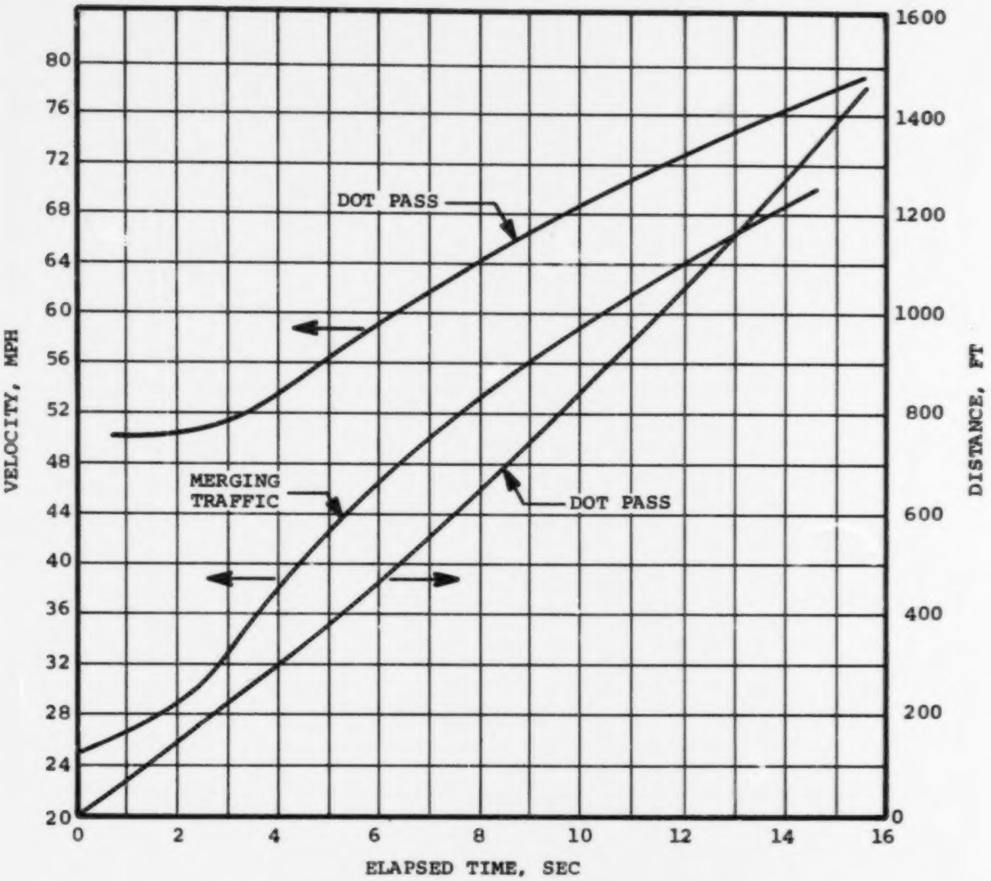
1. AMBIENT TEMPERATURE = 105°F
2. ENGINE RATED AT 175 HP (59°F DAY)
3. ACCESSORY LOAD = 4 HP
4. 4000-LB VEHICLE



STANDING-START VEHICLE ACCELERATION PERFORMANCE WITH FREE-TURBINE REGENERATED ENGINE

FIGURE 7-9

NOTES:
1. AMBIENT TEMPERATURE = 150°F
2. ENGINE RATED AT 175 HP (59°F DAY)
3. ACCESSORY LOAD = 4 HP
4. 4000-LB VEHICLE



MERGING TRAFFIC AND DOT HIGH-SPEED PASS PERFORMANCE WITH FREE-TURBINE REGENERATED ENGINE

FIGURE 7-10

TABLE 7-6

FUEL ECONOMY OF SINGLE-SHAFT REGENERATED ENGINE (4000-LB VEHICLE, 4-HP ACCESSORY LOAD, BELT TRANSMISSION)

Rated Power Level, 59°F Without Boost, shp	Ambient Temperature, °F	Fuel Consumption, mpg (4-hp Accessory Load)							Composite USEDC*	
		FDC (1.3-hp Accessory load)	20 mph	30 mph	40 mph	50 mph	60 mph	70 mph	mpg	Btu/mile
108	105	16.8	25.5	30.1	30.0	26.7	23.1	20.8	21.2	5450
108	85	17.0	25.2	30.4	30.8	27.4	24.3	20.5	22.1	5240
108[a]	85[a]	15.8	24.7	29.0	29.8	26.3	22.0	19.9	20.8	5550
108[b]	85	16.2	26.1	29.9	31.1	27.6	22.5	18.9	21.2	5450
108	59	17.2	26.5	31.5	31.7	23.2	24.3	22.0	22.7	5100
108	30	17.3	27.0	32.1	32.6	30.7	25.2	21.2	23.0	5030
122[c]	85	14.6	23.1	26.6	27.0	23.8	21.6	18.0	19.3	6000
155	85	14.0	21.0	26.3	28.4	27.2	24.7	20.8	19.7	5860
175[d]	59	12.5	16.1	18.2	18.2	17.2	15.5	13.7	14.8	7440
175[e]	59	14.2	20.8	25.6	24.0	21.2	18.7	16.6	18.0	6100

*Appendix 1

[a] Toroidal transmission

[b] Torque converter used instead of slip-clutch

[c] Engine rated at 122 hp, vehicle weight = 4600 lb, traction transmission

[d] Spark ignition engine with three-speed automatic transmission - 4300-lb vehicle

[e] Spark ignition engine - traction transmissic

TABLE 7-7

FUEL ECONOMY OF SINGLE-SHAFT RECUPERATED ENGINE
(4000-LB VEHICLE, 4-HP ACCESSORY LOAD, BELT TRANSMISSION)

Rated Power Level, 59°F Without Boost, shp	Ambient Temperature, °F	Fuel Consumption, mpg							Composite USEDC[*]	
		FDC (1.3-hp Accessory load)	20 mph	30 mph	40 mph	50 mph	60 mph	70 mph	mpg	Btu/mile
125	105	14.2	20.3	25.9	28.1	27.7	24.2	20.8	19.7	5870
125	85	14.3	19.9	25.7	28.2	27.5	25.3	22.1	20.0	5780
125	59	14.2	20.6	26.2	28.5	27.9	25.0	21.7	20.0	5780
125	30	14.6	20.6	26.1	28.5	28.2	25.6	22.3	20.3	5700
175[a]	59	12.5	16.1	18.2	18.2	17.2	15.5	13.7	14.8	7440
175[b]	59	14.2	20.8	25.6	24.0	21.2	18.7	16.6	18.0	6100
125[c]	85	11.5	20.1	24.8	26.4	24.7	22.2	19.2	17.2	6730

[*] Appendix 1

[a] Spark ignition engine with three-speed automatic transmission - 4300-lb vehicle

[b] Spark ignition engine - traction transmission

[c] Traction transmission

TABLE 7-8

FUEL ECONOMY OF FREE-TURBINE REGENERATED CYCLE
(4000-LB VEHICLE, 4-HP ACCESSORY LOAD, THREE-SPEED AUTOMATIC TRANSMISSION

Rated Power Level, 59°F Without Boost, shp	Ambient Temperature, °F	Fuel Consumption, mpg							Composite USEDC*	
		FDC (1.3-hp Accessory load)	20 mph	30 mph	40 mph	50 mph	60 mph	70 mph	mpg	Btu/mile
175	85	16.4	20.6	23.2	23.3	23.4	22.3	20.6	19.9	5810
175[a]	85	15.8	19.1	24.1	22.0	21.8	20.6	18.9	18.9	6130
135	85	18.1	23.3	23.8	25.9	25.8	24.3	21.1	21.6	5350
135[a]	85	17.4	24.7	23.2	24.2	23.7	22.2	18.9	20.5	5650
175[b]	59	12.5	16.1	18.2	18.2	17.2	15.5	13.7	14.8	7440
175[c]	59	14.2	20.8	25.6	24.0	21.2	18.7	16.6	18.0	6100

*Appendix 1

[a] Traction transmission

[b] Spark ignition engine with three-speed automatic transmission - 4300-lb vehicle

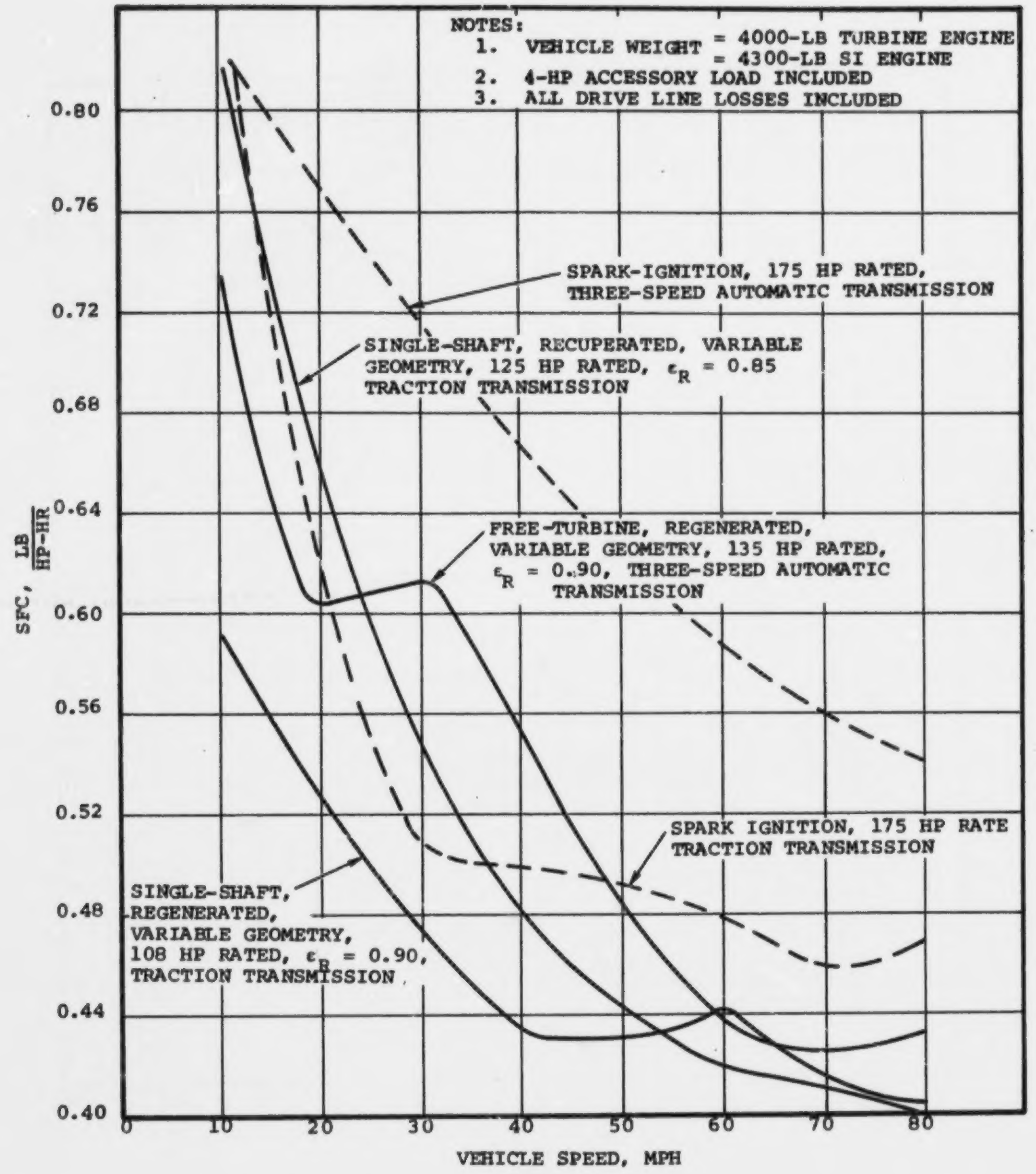
[c] Spark ignition engine - traction transmission

Note in Table 7-6 that the difference in fuel economy between the use of the toroidal and the belt transmission is only about 1 mpg on the FDC and a similar difference for the composite route, even though there are considerable differences between transmission efficiencies (Figures 2-25 through 2-27). The reason for this is that at part-power, the more inefficient toroidal transmission forces the engine to operate at a higher power level where the sfc is lower, thus causing relatively little change in the required fuel flow, for operation at the same road-load with the belt transmission.

Specific fuel consumption, as a function of road-load power requirement, is shown in Figure 7-11 for the three candidate gas turbine engines and the spark-ignition engine. The SI engine is shown for two types of transmissions--the conventional three-speed automatic and the traction drive. Constant speed fuel economy for the candidate gas turbine and the SI engine is illustrated in Figure 7-12.

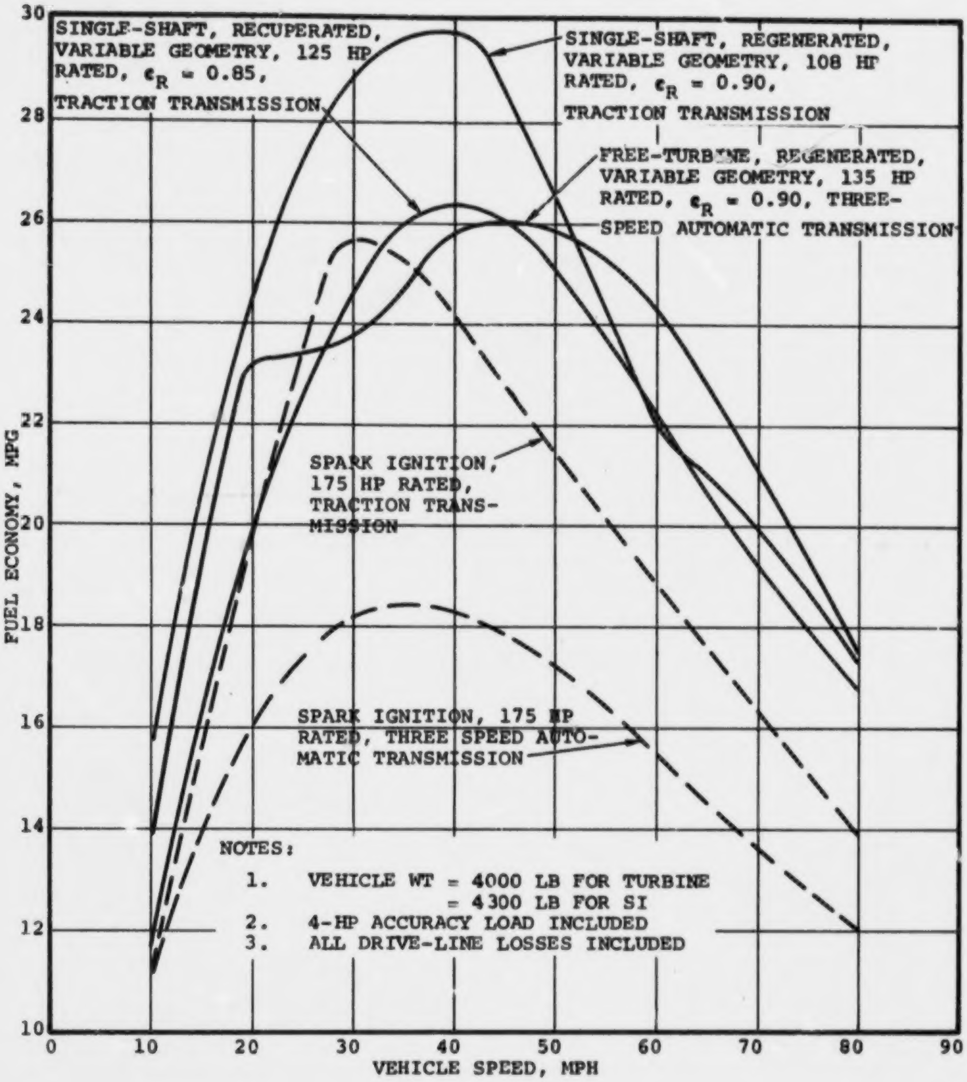
Note that the conventional three-speed automatic transmission is used with the free-turbine engine in these comparisons because of greater fuel economy for the FDC at constant speeds (Table 7-8). This result is illustrated in Figure 7-13 by a comparison of the constant-speed fuel economy curves for the power-boosted free-turbine engine with the traction drive and three-speed automatic transmission. A similar fuel economy comparison for the single-shaft regenerated engine is shown in Figure 7-14.

For the purpose of comparisons, road-load sfc curves for several other engines analyzed in this study are plotted in Figure 7-15. An optimum single-shaft engine is also shown to emphasize the reduction in sfc achieved by optimizing the gas turbine engine design for the automobile application.



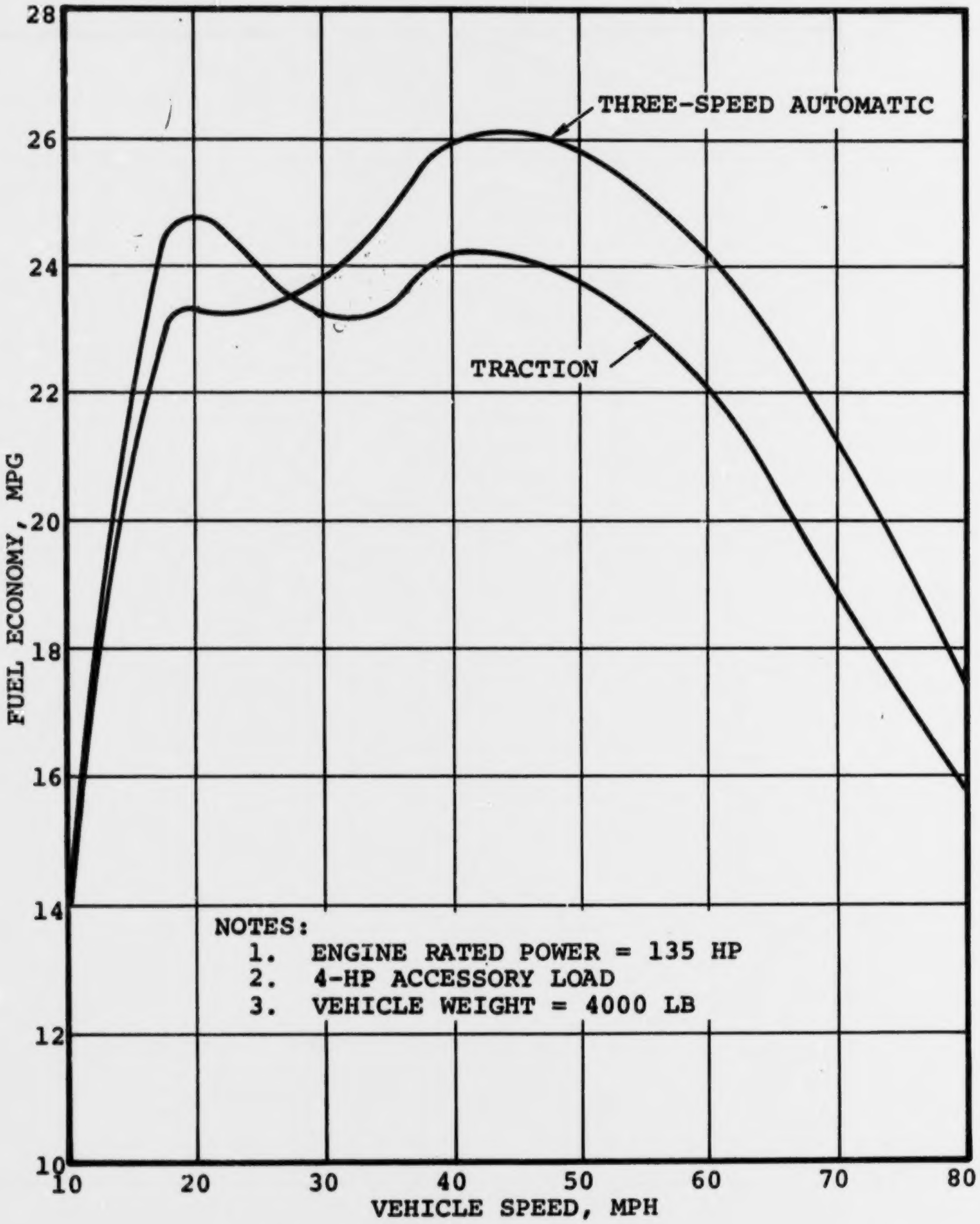
ROAD-LOAD SFC COMPARISONS OF THREE
CANDIDATE TURBINES AND SI ENGINE
85°F SEA-LEVEL DAY

FIGURE 7-11



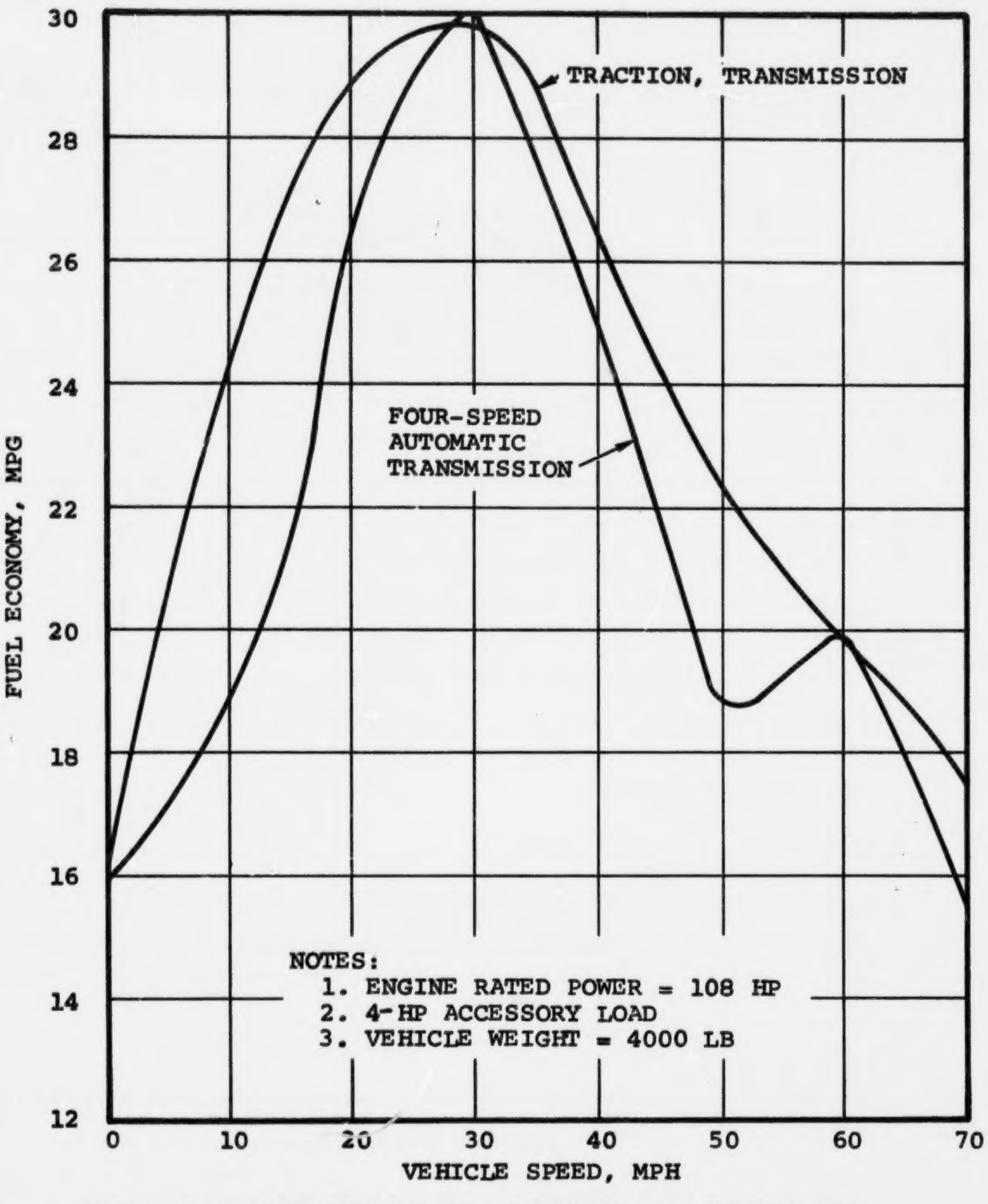
ROAD-LOAD FUEL ECONOMY COMPARISONS OF THREE CANDIDATE ENGINES AND SI ENGINE, 85°F DAY, SEA LEVEL

FIGURE 7-12



ROAD-LOAD FUEL ECONOMY COMPARISON FOR FREE-TURBINE REGENERATED ENGINE WITH VARIABLE GEOMETRY, $\epsilon_R = 0.90$, 85°F SEA-LEVEL DAY

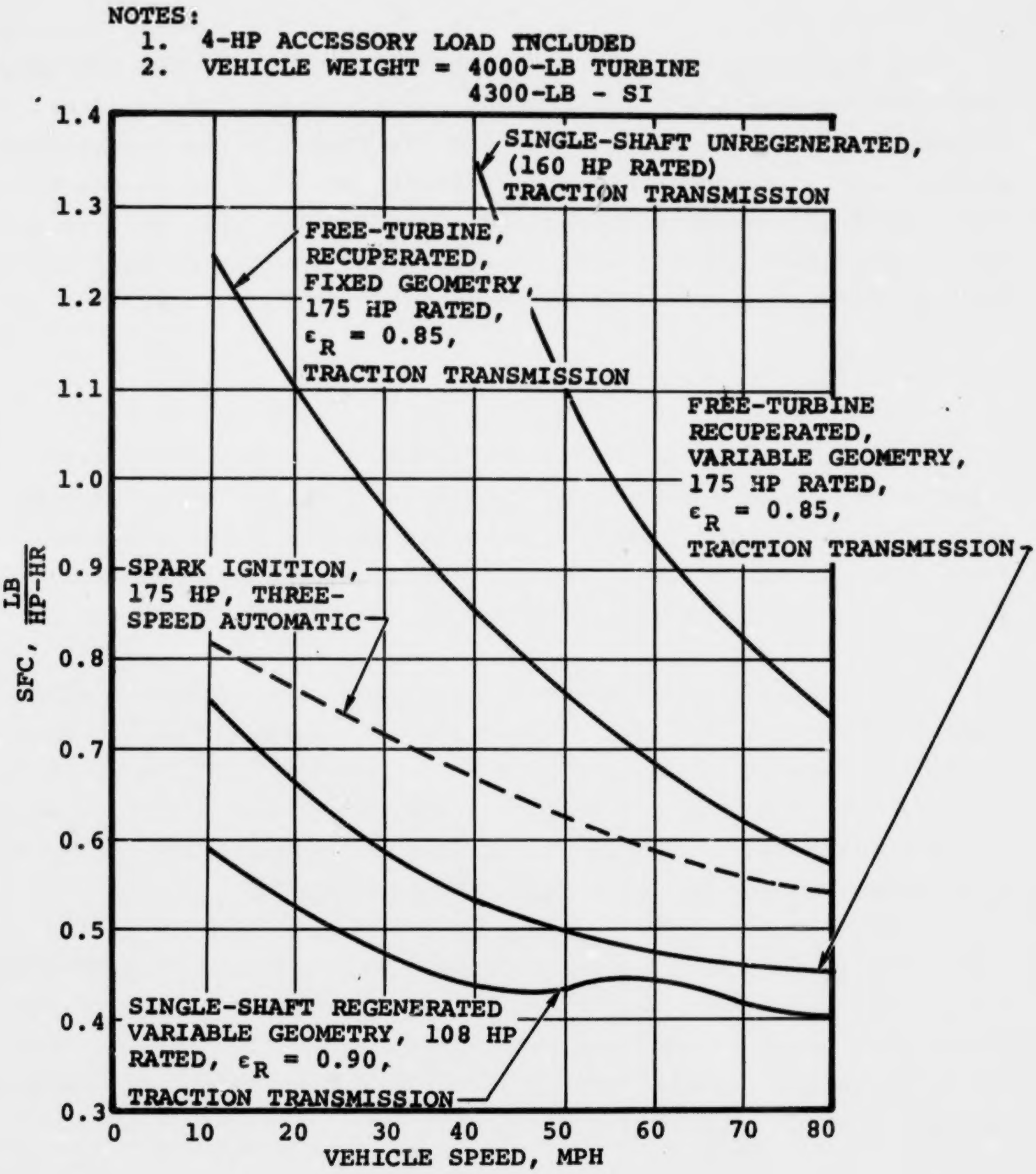
FIGURE 7-13



ROAD-LOAD FUEL ECONOMY OF SINGLE-SHAFT, REGENERATED ENGINE, 85°F SEA-LEVEL DAY

FIGURE 7-14

NOTES:
1. 4-HP ACCESSORY LOAD INCLUDED
2. VEHICLE WEIGHT = 4000-LB TURBINE
4300-LB - SI



ROAD-LOAD SFC COMPRARISONS,
85°F SEA-LEVEL DAY

FIGURE 7-15

The result of the fuel consumption analysis is that the single-shaft regenerated engine with power boost provides the greatest fuel economy for the composite route, and the free-turbine regenerated engine with power boost has the greatest fuel economy on the FDC. The difference between this engine and the spark-ignition unit is about 7 mpg, which means a difference of about $800 in cost of fuel for the life of the vehicle, when the fuel cost is $0.35/gal (Figure 7-1).

7.5 EMISSIONS OF CANDIDATE ENGINES

Emissions data recorded for a number of AiResearch engines provided a preliminary computation procedure for estimating exhaust gas from turbine engines. The main purpose of this procedure was to provide a comparison of the various cycles, on a relative emissions basis, rather than predictions of absolute values for the emissions.

The emissions of greatest concern are carbon monoxide (CO), total unburned hydrocarbons (THC), and oxides of nitrogen (NO_x). Analysis of engine data has shown that the weight ratio of THC to CO is approximately a linear function of combustion efficiency. This relationship has been utilized to provide a prediction procedure for combustors with known combustion efficiency characteristics.

Oxides of nitrogen are not specifically related to combustion efficiency but are characterized by an effective fraction of the emissions that would be obtained with a stoichiometric mixture. Procedures for estimating all three pollutants are discussed in Appendix 4 of this report.

These procedures were implemented in a computer program that accepts the cycle map decks and punches a new deck, including the three emission indexes. This deck is then used in the mission analysis program to integrate total emissions over a specified driving cycle.

Initial cycle computations were performed with the EPA low-pressure ratio regenerated combustor design, using an effective stoichiometric fraction of 0.2 and a THC-CO ratio at a 99-percent efficiency of 0.1205, as discussed in Appendix 3.

Results of the relative emissions calculations for the three candidate cycles are shown on Table 7-9. These are presented as nondimensional ratios, in which each emission level for each engine was ratioed to the largest value for that type of emission.

As might be expected, the lowest NO_x level is obtained for the engine with the lowest regeneration effectiveness. The last two engines in this table--the single-shaft regenerated (108 hp) and the single-shaft recuperated (125 hp)--appear to be the better cycles with respect to emissions.

Because of the parallel combustor optimization contract with the OAP (No. 68-04-0014) and because the analytical techniques available were not completely verified experimentally, no attempt was made to improve the absolute values of the predictions nor the corresponding relative values shown on Table 7-9. If such an attempt were to be made, the NO_x value for the single-shaft recuperated engine might be reduced by decreasing the combustor volume and, thus, reducing the residence time in the combustor, although, this procedure would probably increase the CO and THC values. However, if another possible solution were used--namely, that of extracting combustor primary zone air upstream from the heat exchanger--the regenerated cycle NO_x emissions would be expected to have a percentage reduction greater than that for the recuperated engine. This effect would result from the lower compressor discharge temperature of the lower pressure ratio regenerated engine. Finally, if combustor variable geometry were used, the comparative and absolute results could be affected in an entirely different manner. Therefore, the results on Table 7-9 are meaningful

TABLE 7-9

EMISSIONS COMPARISON OF CANDIDATE CYCLES FOR FEDERAL DRIVING CYCLE (85°F DAY)

Engine	Rated Power Level, 59°F Without Boost, shp	Regeneration Effectiveness at Maximum Power	Relative Emissions			Average, mpg*
			NO_x	CO	THC	
Free-Turbine Regenerated	175	0.90	1.00	1.00	1.00	16.4
	135	↓	0.99	0.98	0.98	17.0
Single-Shaft Regenerated	155		0.99	0.90	0.79	14.0
	108		0.81	0.69	0.58	17.0
Recuperated	125	0.85	0.70	0.37	0.10	14.3

*Fuel LHV = 18,500 Btu/lb; density = 6.25 lb/gal only for those engines for which the prediction procedure are applicable, based on experimental data obtained on existing aircraft engines, not necessarily optimized for low emissions.

SECTION 8

PRELIMINARY DESIGN AND MANUFACTURING COST ESTIMATES FOR CANDIDATE CYCLES

8.1 ENGINE DESIGN CHARACTERISTICS

The general approach taken for each of the engine layouts was to make the design as simple as possible, keeping in mind that automated procedures must be used in manufacturing every part. Complex shapes and costly machined parts were minimized where possible.

Each configuration features a single-stage centrifugal compressor with continuously variable inlet guide vanes. Aluminum was chosen for most compressors, except the higher pressure ratio recuperated engine where 17-4 pH was used because of the higher discharge temperature.

A single-can combustor with a vaporizer fuel injector was selected to provide the lowest system cost and possibly the lowest emissions for each engine configuration.

A single-stage radial turbine was used for each single-shaft engine and the gas generator stage of the free-turbine engine. An axial power turbine stage was employed in the latter unit.

The maximum turbine inlet temperature was limited to 1900°F for both single-shaft engines, so that the desired 200-hr life could be achieved with a cast turbine wheel without rotor cooling. For the free-turbine engine, the turbine inlet temperature was limited to 1850°F to achieve the required life without cooling, since the relative blade temperature is higher due to the lower expansion ratio across this first-stage of the turbine.

Engine life at maximum power was estimated in the required range of 200 hr for each of the regenerated candidate engines. The recuperated turbine life is somewhat marginal because of the high tip-speed required for the higher pressure ratio engine.

To achieve the 200-hr life at maximum power, the pressure ratio would have to be reduced, thus reducing the maximum engine speed, or the maximum turbine inlet temperature could be slightly reduced to obtain the required life.

Each engine was mounted on foil-type gas bearings to eliminate the oil lubrication system normally provided on gas turbines. Some of the advantages of the gas bearing system are:

(a) Reduction in weight and volume

(b) Elimination of power demands and complexity for oil pumping and scavenging

(c) Elimination of oil cooling, defoaming, and filtering requirements and associated installation and maintenance problems

(d) Reduced requiremenents for servicing and accessibility

(e) Reduced initial cost and cost-of-ownership

(f) Elimination of bearing temperature restrictions due to the oil

Inlet air filtration can be achieved through a conventional paper filter with a rated flow of about twice the maximum inlet airflow, to allow for severe dust conditions. Commercially available filters can remove particles to 2-3 microns with a clean restriction of about 0.4 in. of water.

Water injection is accomplished by spray jets at the compressor inlet. The water is not expected to have any harmful effects on the engine, provided de-ionized water is used. A suitable chemical agent might be added to the water periodically to remove the road film collected on static and rotating parts.

Significant engine design characteristics for the candidate cycles are shown on Table 8-1. All performance numbers in this table are values computed for the 100-percent power design-point.

8.1.1 Rotary Regenerators

Rotary regenerator sizing studies were conducted to aid in the selection of engine design-point conditions. The glass-ceramic matrix, employed in this investigation, is assumed to have the following properties:

Surface density, ft^2/ft^3 = 1728

Porosity = 0.72

Passage hydraulic diameter, in. = 0.02

Matrix web thickness, in. = 0.00357

Material density, lb/ft^3 = 171

These matrix properties are target values proposed by the Corning Glass Works for "J" process Cercor discs. The required disc diameter

TABLE 8-1

CANDIDATE ENGINE DESIGN CHARACTERISTICS*

Components	Unit	Configuration			
		Single Shaft	Free Turbine	Single Shaft	Single Shaft
Compressor					
Inlet Temperature	°R	518.7	518.7	518.7	518.7
Airflow	lb/sec^2	1.748	2.106	1.218	1.207
Pressure Ratio		4.607	4.558	4.607	6.415
Efficiency	%	0.822	0.813	0.811	0.790
Speed	rpm	70,228	67,500	84,100	83,600
Impeller Diameter	in.	5.551	5.94	4.634	5.26
Polar Moment of Inertia	$lb\text{-}in.\text{-}sec^2$	0.0064	0.00646	0.0018	0.00721
Materials		Aluminum			17-4 pH
Combustor					
Inlet Temperature	°R	1690	1660.0	1690.0	1571.4
Efficiency	%	99.9	99.90	99.90	99.9
Compressor Turbine					
Inlet Temperature	°F	1900	1850.0	1900.0	1900.0
Pressure Ratio		3.659	2.026	3.659	5.092
Corrected Airflow	lb/sec	0.828	1.001	0.577	0.435
Efficiency	%	0.890	0.880	0.890	0.875
Rotor Diameter	in.	6.145	5.55	5.13	6.025
Polar Moment of Inertia	$lb\text{-}in.\text{-}sec^2$	0.0254	0.0152	0.0103	0.0228
Material		INCO 713C			
Blade Inducer Stress	ksi	50-60	43	53	67
Blade Exducer Stress	ksi	35-45	30-45	30-44	50-65
Disc Average Tangential Stress	ksi	45-55	43	50	82
Minimum Life	hr	175-225	125-175	175-225	50-100
Power Turbine					
Inlet Temperature	°F		1516.0		
Pressure Ratio			1.769		
Corrected Airflow	lb/sec		1.956		
Efficiency	%		0.88		
Rotor Diameter	in.		6.21		
Polar Moment of Inertia	$lb\text{-}in.\text{-}sec^2$		0.0192		
Material					
Speed	rpm		55,200		
Blade Root Stress	ksi		66		
Disc Average Tangential Stress	ksi		55		
Blade Life	hr		50-100		
Heat Exchanger					
Type		Rotary Regenerator			Counterflow Recuperator
Effectiveness		0.90	0.90	0.90	0.85
Material		Ceramic			
Leakage	%	5.4	5.4	5.4	0.0
Performance					
Specific Fuel Consumption	lb/hp-hr	0.41	0.423	0.410	0.434
Power	shp	155.0	175.0	108.0	125.0

*All performance numbers are values for the 100% power, design-point. See Section 2 for assumptions used in these calculations.

and thickness were determined as a function of effectiveness and fractional pressure drop by a computer program, using experimental friction factors and heat-transfer data supplied by Corning.

To illustrate the results of this investigation, regenerator sizing data are depicted for the single-shaft engine. The nominal air- and gas-side conditions are listed below:

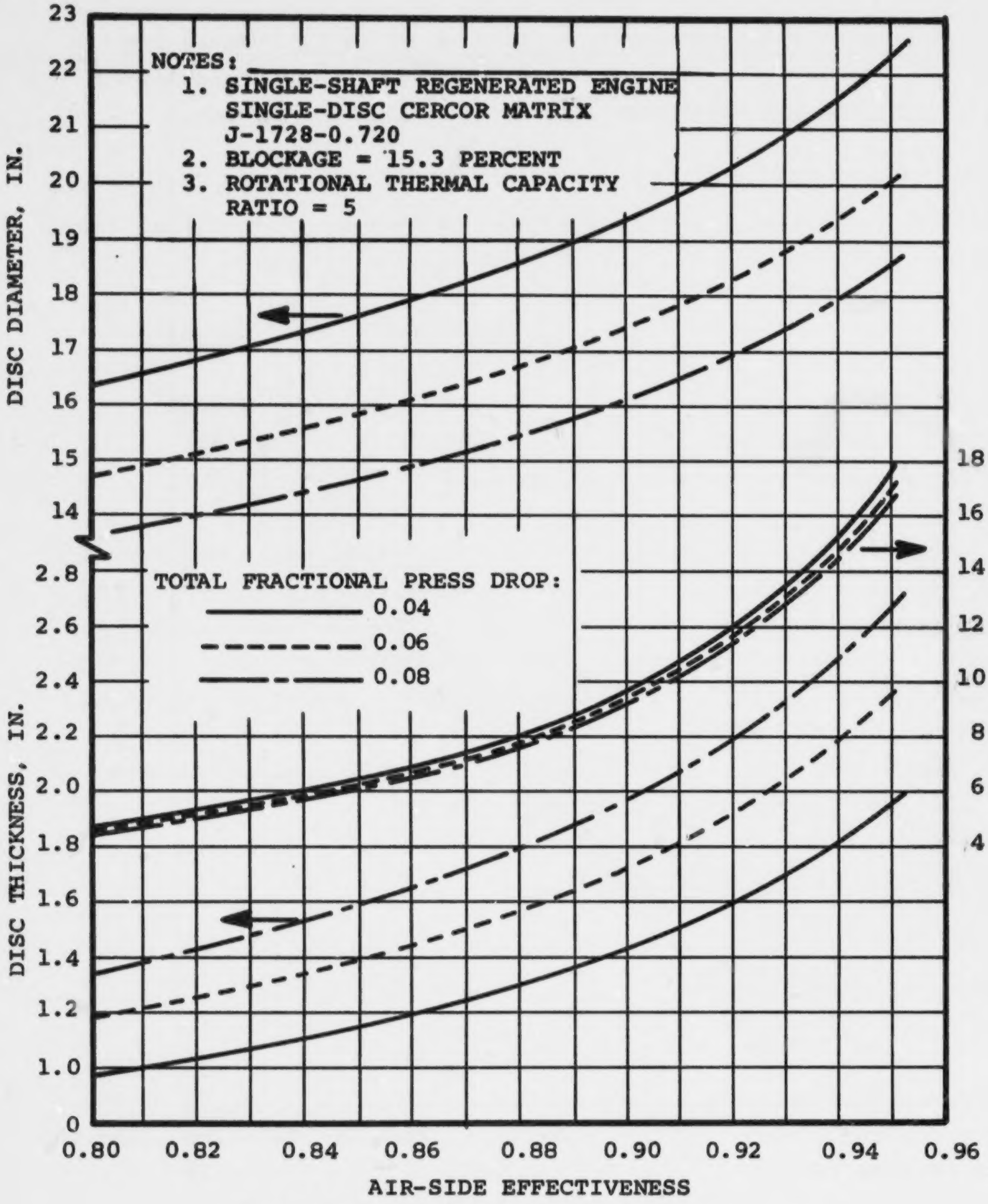
	Air Side	Gas Side
Flow rate, lb/sec	1.129	1.225
Inlet temperature, °R	861	1781
Inlet pressure, psia	66.0	15.91

Disc diameter, thickness, and weight are plotted versus air-side effectiveness for a single-disc regenerator in Figure 8-1, assuming 15.3-percent blockage by the seals and hub area. Equal air- and gas-side areas are assumed. Similar data are plotted in Figure 8-2 for a twin-disc design having 22.3-percent blockage. Percent of blockage figures are tentative and are the result of an earlier AiResearch study. Disc rotational speeds vary from 20 to 60 rpm, as required, to satisfy the rotational thermal capacity ratio.

8.1.2 Fixed-Boundary Recuperators

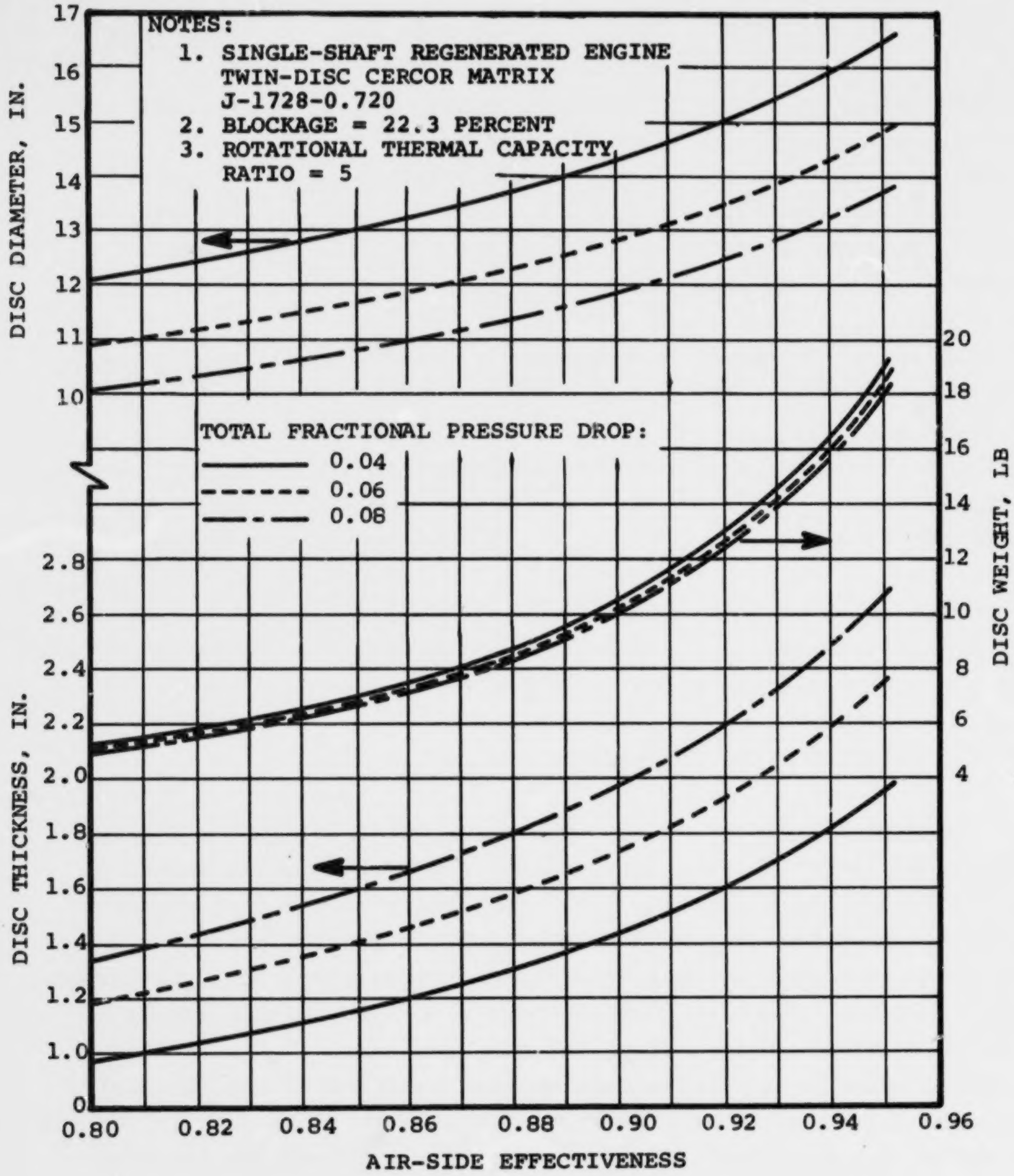
The metal plate-fin matrix, assumed in the calculations, has the following characteristics:

	Air Side	Gas Side
Surface Designation	20R-.100-.125-.004	16R-.153-.143-.004
Number of fins per in.	20	16
Fin height, in.	0.100	0.153



CERCOR REGENERATOR SIZING CHARACTERISTICS

FIGURE 8-1



CERCOR REGENERATOR SIZING CHARACTERISTICS

FIGURE 8-2

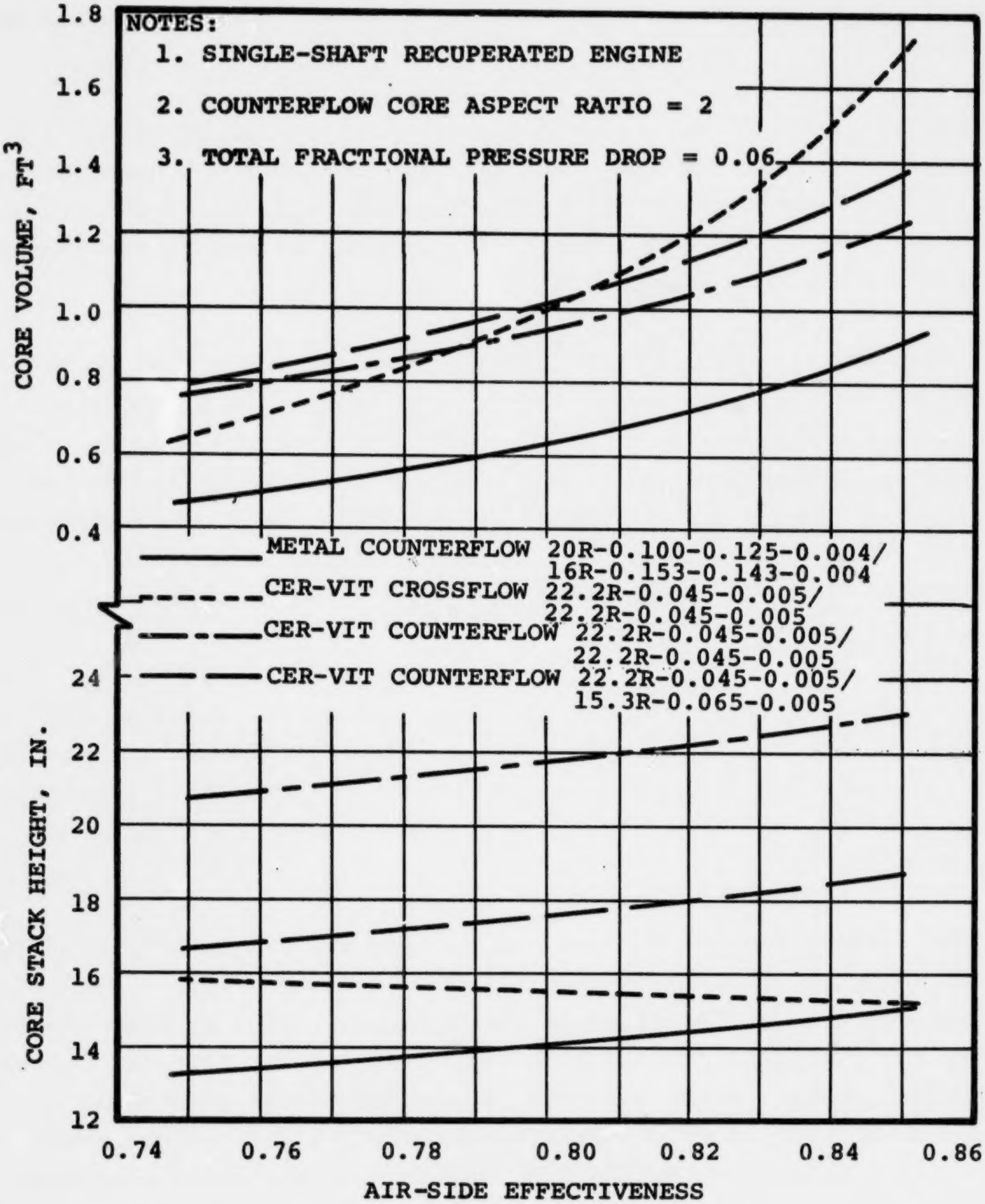
	Air Side	Gas Side
Fin thickness, in.	0.004	0.004
Surface density, ft^2/ft^3	250	292
Hydraulic radius, ft	0.001295	0.00175
Fin area ÷ surface area	0.676	0.718
Material density, lb/ft^3	494	494
Separator plate thickness, in.	0.01	0.01

The Cer-Vit matrixes were assumed to have a square flow passage, with a fin pitch of 22.2/in. and a fin thickness of 0.005 in. The actual passage heights were varied in the study, since these can be readily tailored to meet customer requirements. Cer-Vit has a density of approximately 156 lb/ft^3 and can be fabricated in a wide range of shapes and sizes.

Comparisons of core stack height and volume for counterflow metallic, counterflow ceramic, and crossflow ceramic recuperators for the single-shaft engine are shown in Figure 8-3.

8.2 MANUFACTURING COST ESTIMATING PROCEDURE

The manufacturing plan for producing a gas turbine engine and power transmission for the conventional family automobile would, of necessity, include a facility with equipment optimized for the technology required for the uniqueness of the gas turbine product. The plan visualized is a concept of advanced total automation with computerized controls that have complete control of processing, starting with raw material through final assembly and test. The automated equipment would be a continuous-flow type tailored to the distinct features of the gas turbine engine, such as thin cross sections of the air foils, super alloys, and high accuracies required for the high-speed rotating components. Under these conditions, the cost of



COMPARISON OF RECUPERATOR CONFIGURATIONS

FIGURE 8-3

producing the hardware would tend to approach the material costs, with direct labor accounting for only a small portion of total manufacturing cost.

These optimized automated facilities are assumed to exist, in determining the material and manufacturing labor costs. However, no estimate was made of the capital expenditures required to achieve the facilities; therefore, the costs do not include allowance for amortization of non-recurring expenses.

Raw materials, like the facilities, must be controlled to forms compatible with the processes, to minimize handling and stock-removal operations. Material costs include all those incurred in delivering the material ready for final processing, such as extruding, rolling, machining, or forming. As an example, material costs of castings and forgings include the primary metal plus the labor involved in the process so that the only remaining direct cost is the manufacturing labor required for final machining.

Estimates of casting material costs include the salvage value incurred by recycling excessive casting material inherent to this process. However, the final cost of the part is not adjusted for the reclaimable value of scrap, such as heavy meltings, turnings, and stamping clippings.

The costs estimated for the gas turbine engines in this analysis are considerably lower than those obtained from the present aircraft-type engines and extrapolating to large volumes. The primary reason is that the engines discussed here were designed for low cost rather than light weight. Where possible, the parts of the automobile engines were designed as castable rather than as lightweight sheet metal. As a result of the low cost design, the engine is much heavier than a comparable aircraft engine but is considerably lighter than the comparable spark-ignition engine.

The manufacturing costs of producing the hardware were established by an economic plan typical of a normal transportation equipment manufacturer.

Preliminary make-or-buy decisions were made to determine the type or mix of hardware that would be best suited for manufacturing "in-house" or procuring from an outside vendor--the lowest cost being the most important criterion. The ultimate decision would have to be made on the basis of the final design from that manufacturer electing to produce this type of equipment. For purposes of this report, a decision to purchase a part will be indicated in the estimate breakdown for each proposed system. The purchasing plan will be one of placing managed contracts with competitively selected vendors on an annual basis to obtain the lowest cost.

In computing manufacturing costs, the direct labor rate was obtained from the October 7, 1971, issue of the Monthly Labor Review[10] for transportation equipment workers. The manufacturing labor cost was determined as follows:

Direct labor	$ 4.40/hr
Fringe benefits (approx. 35 percent)	1.60/hr
Manufacturing overhead	4.00/hr
Manufacturing labor cost	$10.00/hr

Manufacturing overheat cost is an estimated number, since it includes many variables that are dependent upon the type and location of the facility and the accounting procedures of individual companies or corporations. In general, this information is treated as highly proprietary and, for competitive reasons, is not divulged. Because of the unknowns in manufacturing overhead costs and potential future escalations above the 1971 level, the final cost numbers would have to be adjusted before the initial production date.

Estimates were based on producing 1 million engines per year of each concept, as depicted by the engineering conceptual layout. Each individual detail was analyzed for cost by taking into consideration material cost, form, weight, and complexity, with the production rate for the controlled, single-purpose equipment. An example typical of the logic used in determining manufacturing costs is given in detail for a compressor impeller as follows:

Compressor Impeller - Radial Flow Single-Shaft Engine, Recuperated

Material	17-4 pH
Finished machine weight, lb	1.8
Finished cast weight, lb	2.0
Gating, lb	10.0
Pour weight, lb	12.0

Casting Process	Manufacturing Time, hr	Dollars
Wax molding	0.100	1.00
Assembly	0.025	0.25
Investing	0.245	2.45
Dewax and insulation	0.040	0.40
Casting - pouring	0.060	0.60
Knockout	0.030	0.30
Cut-off snag and bend	0.125	1.25
First inspection (visual and zyglo)	0.100	1.00
	0.725	7.25
Material cost		4.00
		11.25
Yield, 85 percent		13.23
Final inspection (x-ray and dimen.)	0.250	2.50
		15.73
Yield, 90 percent		17.47
Marking and master heat qual.	0.050	0.50
TOTAL		17.97

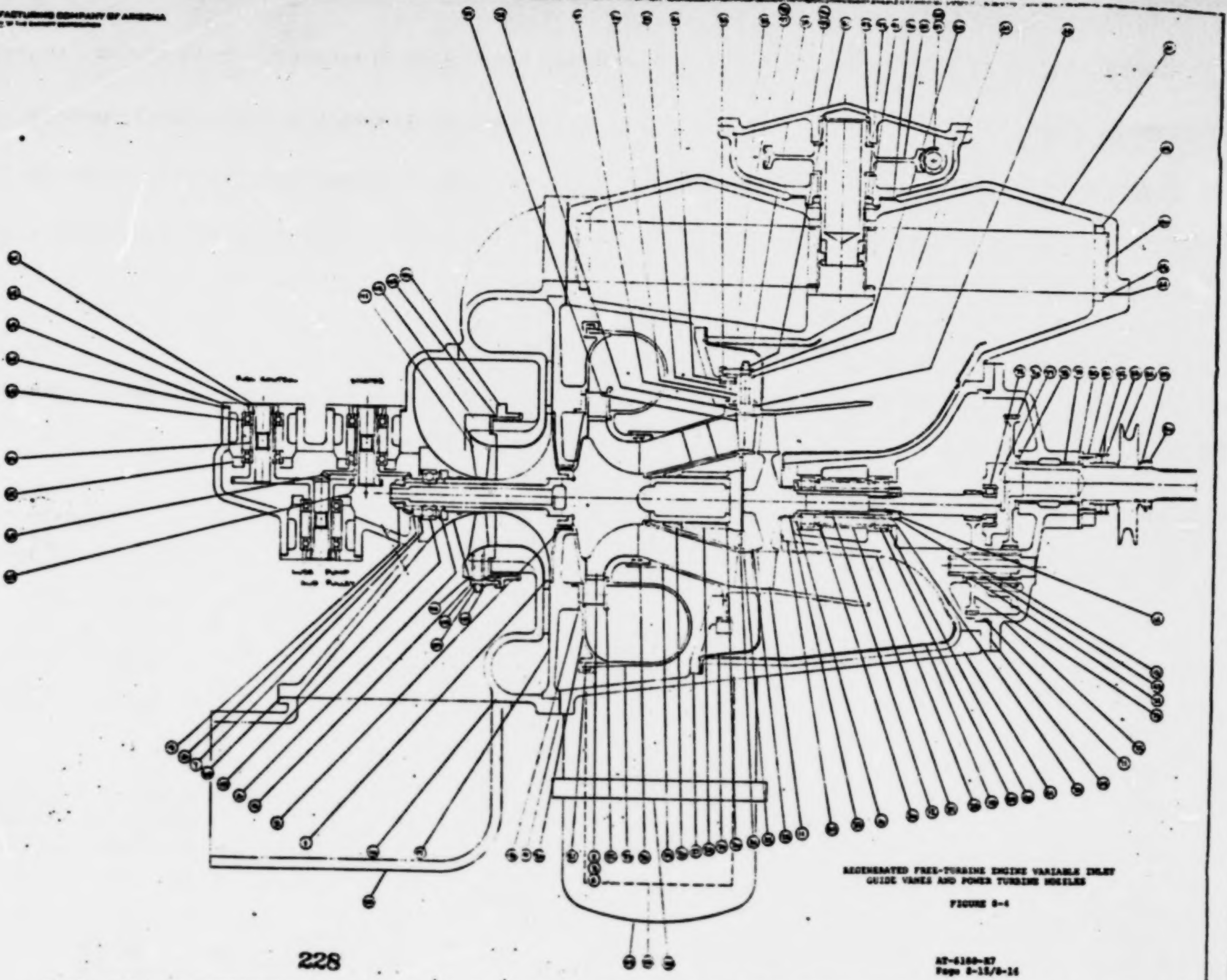
Machining Process	Manufacturing Time, hr	Dollars
Contour turn aft end, bearing diam, and large outside diam	0.050	0.50
Face end, spot drill, drill and countersink aft end	0.050	0.50
Face, spot drill, bottom drill, and countersink front end	0.050	0.50
Grind large outside diam and tip shroud	0.035	0.35
Grind bearing diam and end face	0.030	0.30
Grind inside diameters	0.041	0.41
Deburr	0.016	0.16
Shot peen	0.025	0.25
Balance	0.083	0.83
	0.380	3.80
Material cost - casting		18.00
Manufacturing cost		3.80
TOTAL		21.80

Manufacturing costs of purchased engine and transmission parts (such as bearings, seals, the regenerator drive motor, hydraulic clutch, and similar parts) were obtained from manufacturers. (A list of manufacturers consulted is given in Section 13.) Manufacturing costs of vehicle accessories (such as the alternator, battery, starter, etc.) were obtained through automotive marketing personnel.

8.3 DESIGN LAYOUTS AND COST ESTIMATES

Design layout drawings of the candidate engines and transmissions are shown in Figures 8-4 through 8-9. Detailed parts lists, including estimated manufacturing costs, follow each layout (Tables 8-2 through 8-7, respectively).

Control system costs and weights for two types of systems, the hydromechanical and electronic, are estimated on Tables 8-8 and 8-9.



REGENERATED FREE-TURBINE ENGINE VARIABLE INLET GUIDE VANES AND POWER TURBINE NOZZLES

FIGURE 8-4

TABLE 8-2
REGENERATED FREE-TURBINE ENGINE

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
1	Impeller, Compressor	InvCst	AL	1	1.00	5.00	0.15	1.50	6.50
2	Wheel, Turbine	InvCst	713	1	4.11	32.00	0.40	4.00	36.00
3	Shaft, Impeller, Comp	Bar	17-4	1	0.56	2.80	0.01	0.10	2.90
4	Shaft, Impeller, Turbine	Tubing	A286	1	0.77	1.23	0.01	0.10	1.33
5	Seal, Labyrinth	Tubing	440	1	0.20	0.11	0.055	0.55	0.66
6	Slinger, Oil	PwrMtl	4600	2	0.04	0.25	0.02	0.20	0.45
7	Retainer, Bearing	PwrMtl	4600	1	0.01	0.20	0.01	0.10	0.30
8	Gear, Bevel, Spiral	Bar	4620	1	0.35	0.06	0.32	3.20	3.26
9	Nut	Bar	4130	1	0.15	0.03	0.015	0.15	0.18
10	Bearing, Ball	Purch	M-50	3	0.45	18.00	--	--	18.00
11	Wheel, Turbine	InvCst	713	1	2.33	35.00	0.17	1.70	36.70
12	Spacer, Bearing	Tubing	1141	1	0.15	0.01	0.015	0.15	0.16
13	Retainer, Nut	Sheet	4130	1	--	0.01	0.003	0.03	0.04
14	Nut	Bar	4130	1	0.02	0.01	0.01	0.10	0.11
15	Carrier, Bearing	InvCst	4130	1	0.70	4.00	0.04	0.40	4.40
16	Scroll, Compressor	SndCst	GM-60	1	24.00	15.00	0.15	1.50	16.50
17	Diffuser	InvCst	GM-60	1	15.00	13.60	0.25	2.50	16.10
18	Spacer	SndCst	Ni Rst	1	0.57	1.50	0.03	0.30	1.80
19	Seal	Purch	AsbCmp	1	--	0.15	--	--	0.15
20	Shroud	InvCst	310S	1	1.55	12.00	0.05	0.50	12.50
21	Coupler, Scroll	Forgng	310S	1	0.75	7.00	0.08	0.80	7.80
22	Inlet, Impeller, Turbine	InvCst	713	1	0.62	20.00	0.12	1.20	21.20
23	Scroll, Inlet, Turbine	SndCst	310	1	6.00	30.00	0.10	1.00	31.00
24	Seal	Purch	Steel	2	0.10	0.30	--	--	0.30

TABLE 8-2 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
25	Housing, Stator	InvCst	HS-31	1	5.25	100.00	0.25	2.50	102.50
26	Carrier, Bearing, Turbine	InvCst	HS-31	1	1.63	12.00	0.10	1.00	13.00
27	Support, Housing, Stator	SndCst	GM-60	1	6.60	6.00	0.08	0.80	6.80
28	Bearing Assembly, Foil	PwrMtl	316L	1	0.25	2.70	0.22	2.20	4.90
29	Shroud	Sheet	321	2	0.60	0.90	0.01	0.10	1.00
30	Ring, Retainer, Bearing	Stamp	Steel	1	--	0.03	--	--	0.03
31	Ring	Stamp	Steel	1	--	0.05	--	--	0.05
32	Ring, Retainer	Stamp	Steel	1	--	0.05	0.02	0.20	0.25
33	Ring, Stator	InvCst	HS-31	1	0.45	6.00	0.05	0.50	6.50
34	Housing, Seal	Forgng	A286	1	0.25	1.60	0.035	0.35	1.95
35	Seal, Turbine	Purch	Cbn/St	2	--	3.00	--	--	3.00
36	Spring, Compressor	Wind	Steel	1	0.10	0.06	--	--	0.06
37	Carrier, Bearing	Tubing	4130	2	0.45	0.10	0.06	0.60	0.70
38	Shroud, Discharge, Turbine	Sheet	321	1	6.00	5.75	0.025	0.25	6.00
39	Shroud, Discharge, Turbine	Sheet	321	1		--	0.25	2.50	2.50
40	Ring, Spacer	Tubing	4130	1	0.02	0.04	0.015	0.15	0.19
41	Plenum, Turbine	SndCst	GM-60	1	62.21	37.20	0.42	4.20	41.40
42	Housing, Bearing, Carrier	SndCst	GM-60	1	12.37	10.00	0.15	1.50	11.50
43	Ring, Retainer	Tubing	4130	1	0.20	0.03	0.015	0.15	0.18
44	Seal	Purch	AshCmp	1	--	0.20	--	--	0.20
45	Seal, Rotary, Regenerator	Purch	Stl/Ct	2	3.56	10.00	--	--	10.00
46	Rotary, Regenerator	Purch	Cramic	1	12.00	37.00	--	--	37.00
47	Housing, Regenerator	SndCst	GM-60	1	30.11	21.00	0.22	2.20	23.20
48	Washer	Sheet	300	30	0.24	0.60	0.004	0.04	0.64

TABLE 8-2 (Contd)

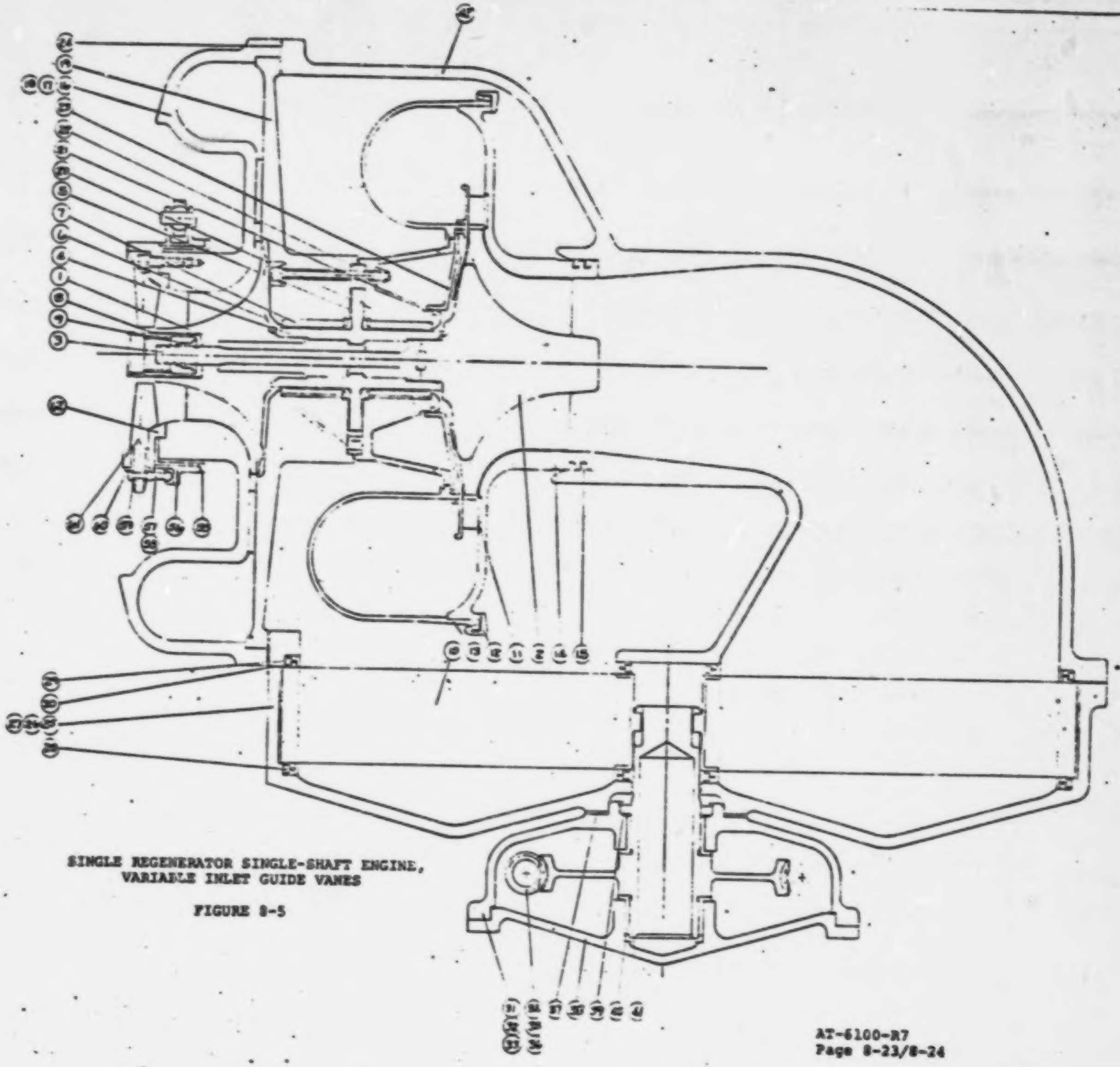
FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
49	Washer, Spring	Stamp	Steel	30	0.06	0.45	--	--	0.45
50	Pin	Bar	Steel	30	--	0.45	--	--	0.45
51	Nut, Self-Locking	Bar	A286	30	0.15	1.20	--	--	1.20
52	Link, Actuator	InvCst	17-4	30	0.21	15.00	0.60	6.00	21.00
53	Ring, Actuator	InvCst	A286	1	0.78	9.00	0.60	6.00	15.00
54	Bearing, Sleeve	Rld/Wld	HS-25	1	0.28	1.40	0.005	0.05	1.45
55	Ring, Support, Actuator	Extrsn	21-6-9	1	0.31	1.80	0.025	0.25	2.05
56	Housing, Vane Support	Bar	303	30	3.25	5.70	0.03	0.30	6.00
57	Ring, Retaining	Rld/Wld	A286	1	0.25	0.80	0.02	0.20	1.00
58	Bearing, Sleeve	Tubing	440A	30	2.50	2.00	0.035	0.35	2.35
59	Vane, Stator	InvCst	713	30	1.70	75.00	0.60	6.00	81.00
60	Housing, Access Gearbox	SndCst	GM-60	1	6.96	5.60	0.15	1.50	7.10
61	Gear, Spur	Forgng	4620	2	1.20	1.00	0.25	2.50	3.50
62	Nut	Tubing	4130	3	0.60	0.10	0.03	0.30	0.40
63	Carrier, Bearing	SndCst	GM-60	3	2.43	3.00	0.15	1.50	4.50
64	Seal	Purch	AsbCmp	3	--	0.12	--	--	0.12
65	Bearing, Ball	Purch	52100	6	0.54	3.12	--	--	3.12
66	Spacer, Bearing	Tubing	4130	3	0.38	0.15	0.015	0.15	0.30
67	Ring, Retainer	Tubing	4130	3	0.60	0.20	0.015	0.15	0.35
68	Gear, Bevel, Spiral	Forgng	4620	1	0.45	0.45	0.45	4.50	4.95
69	Gear, Pinion, Spur	Bar	4620	1	0.50	0.08	0.12	1.20	1.28

TABLE 8-2 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
70	Shaft, Pinion	Bar	6260	1	0.47	0.09	0.14	1.40	1.49
71	"O" Ring	Mold	Silcne	1	--	0.02	--	--	0.02
72	Housing, Gear	SndCst	GM-60	1	7.38	5.16	0.15	1.50	6.66
73	Gear, Idler	Forgng	6260	1	0.64	0.58	0.25	2.50	3.08
74	Bearing, Sleeve	Tubing	Bronze	2	0.08	0.12	0.03	0.30	0.42
75	Bearing, Roller	Purch	M-50	1	0.13	8.00	--	--	8.00
76	Gear	Bar	6260	1	2.17	0.50	0.40	4.00	4.50
77	Support, Bearing	SndCst	GM-60	1	1.47	1.25	0.10	1.00	2.25
78	Bearing, Sleeve	Tubing	Bronze	1	0.20	0.14	0.015	0.15	0.29
79	Bearing, Sleeve	Tubing	Bronze	1		0.14	0.015	0.15	0.29
80	Gasket	Purch	AsbCmp	1	--	0.03	--	--	0.03
81	Spacer	Tubing	4130	1	0.23	0.08	0.03	0.30	0.38
82	Seal, Lip	Purch		2	0.02	0.30	--	--	0.30
83	Carrier, Seal	Cst	GM-60	1	0.28	0.50	0.04	0.40	0.90
84	Pulley	Purch	Steel	1	1.42	0.65	--	--	0.65
85	Retainer, Nut	Stamp	4130	1	0.14	0.01	0.003	0.030	0.04
86	Nut	Bar	4130	1		0.03	0.015	0.15	0.18
87	Gear, Worm	Purch	Bronze	1	0.25	3.64	--	--	3.64
88	Drive Motor			SEE ACCESSORY LIST					
89	Wheel, Worm Gear	Bar	4340	1	2.83	1.61	0.22	2.20	3.81
90	Gasket	Purch	AsbCmp	1	0.03	0.04	--	--	0.04
91	Seal	Purch		1	0.01	0.15	--	--	0.15
92	Bearing, Sleeve	PwdMtl	Bronze	1	0.16	0.30	0.005	0.05	0.35
93	Bearing, Sleeve	PwdMtl	Bronze	1	0.16	0.30	0.005	0.05	0.35

TABLE 8-2 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
94	--	--	--	--	--	--	--	--	--
95	Cover	InvCst	GM-60	1	3.48	2.70	0.05	0.50	3.20
96	Screw, Machine	Purch	Steel	8	0.26	0.08	--	--	0.08
97	Housing, Regenerative Drive	InvCst	GM-60	1	5.03	3.60	0.085	0.85	4.45
98	Screw Machine	Purch	Steel	6	0.20	0.06	--	--	0.06
99	Ignition - Plug	Purch	--	1	0.15	0.20	--	--	0.20
100	Cover, Quill Drive	Tubing	Al	1	0.10	0.03	0.007	0.0	0.10
101	Accoustic Inlet Duct	Purch	Plstc	1	2.50	3.00	--	--	3.00
102	Combustor	Sheet	1018	1		3.70	0.35	3.50	7.20
103	Combustor, Cover	SndCst	GM-60	1	43.26	18.48	0.11	1.10	19.58
104	Combustor, Clamp	Purch	300S.S.	1		1.50	--	--	1.50
105	Support, Actuator	InvCst	410S/S	1	0.47	2.50	0.035	0.35	2.85
106	Ring, Actuator	InvCst	440	1	0.78	6.50	0.055	0.55	7.05
107	Link, Actuator	InvCst	17-4	12	0.08	2.40	0.24	2.40	4.80
108	Pin, Actuator	Purch	410	12		0.06			0.06
109	Nut, Self-Locking	Bar	Steel	12	0.05	0.12	--	--	0.12
110	Bearing, Vane	Mold	Plastc	12	0.03	0.48	--	--	0.48
111	Vane Inlet	InvCst	410	12	0.27	6.00	0.24	2.40	8.40
112	Ring, Vane Support	SndCst	GM-60	1	1.92	2.00	0.15	1.50	3.50
	Miscellaneous Hardward and Unidentified Detailed Parts				38.15	61.00	--	--	61.00
	Assembly Time						0.50	5.00	5.00
	Totals				339.67	717.24	10.267	102.67	819.71



SINGLE REGENERATOR SINGLE-SHAFT ENGINE,
VARIABLE INLET GUIDE VANES

FIGURE 8-5

TABLE 8-3
SINGLE REGENERATOR SINGLE-SHAFT ENGINE

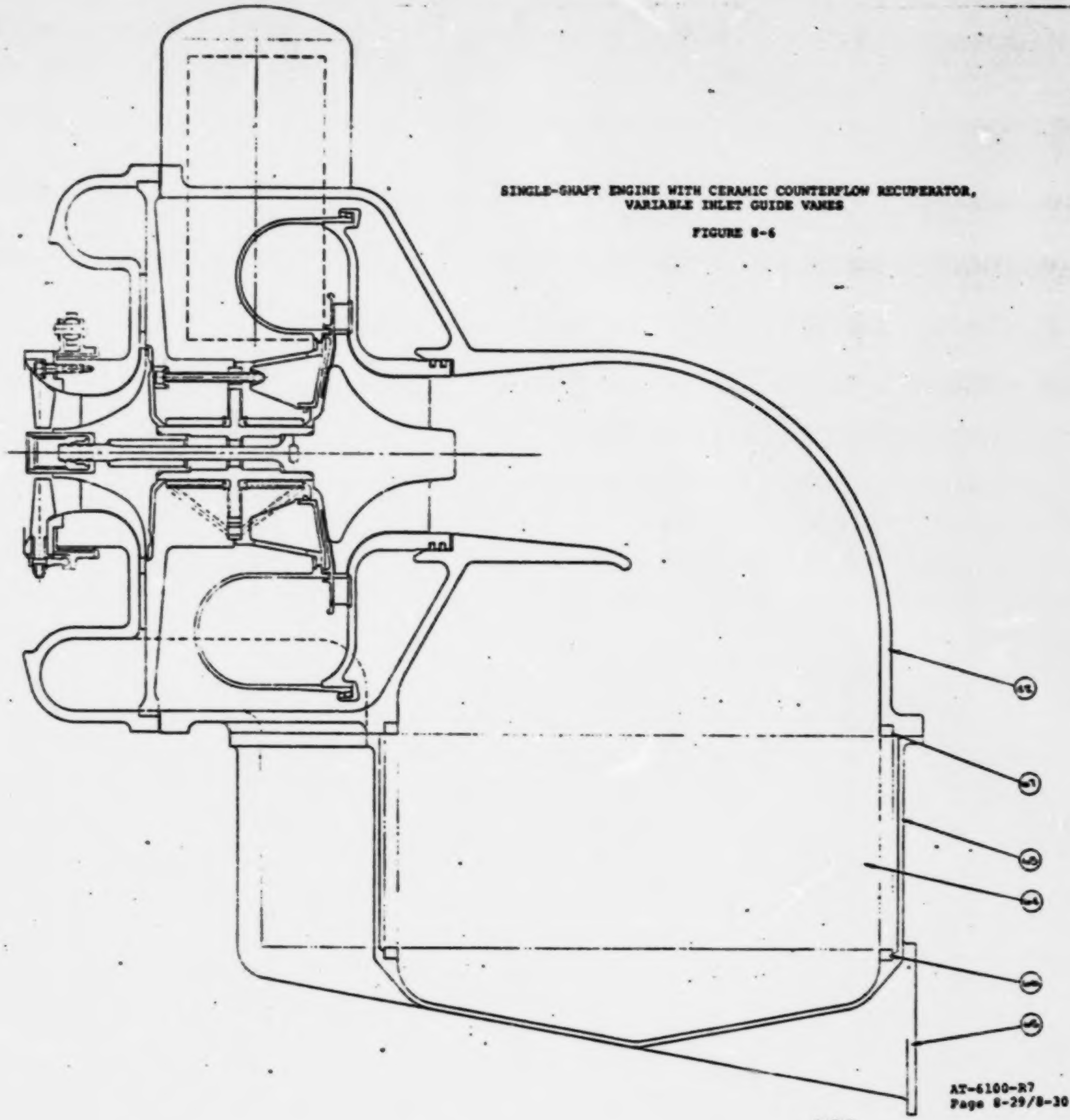
FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
1	Impeller	Inv Cst	Al	1	0.60	3.30	0.15	1.50	4.80
2	Wheel, Turbine	Inv Cst	713C	1	4.11	35.00	0.52	5.20	40.20
3	Shaft, Wheel	Bar	4340	1	0.18	0.34	0.014	0.14	0.48
4	Shaft, Bearing	Forgng	4340	1	1.17	4.00	0.18	1.80	5.80
5	Foil Assembly, Thrust	Sheet	Inc 750	2	0.03	0.30	0.04	0.40	0.70
6	Foil Assembly, Bearing	Sheet	Inc 750	16	0.03	0.30	0.16	1.60	1.90
7	Carrier, Bearing	PwrMet	303	2	0.57	2.40	0.06	0.60	3.00
8	Coupling, Spline	Bar	4340	1	0.08	0.10	0.062	0.62	0.72
9	Nut	Bar	4340	1	0.02	0.03	0.022	0.22	0.25
10	Core, Regenerator	Purch	Cramc	1	11.0	30.00	--	--	30.00
11	Ring, Nozzle	Inv Cst	713C	1	1.23	20.00	0.12	1.20	21.20
12	Ring, Clamp	Rld Rng	310	1	0.94	7.00	0.08	0.80	7.80
13	Scroll, Turbine	Snd Cst	310	1	8.17	30.00	0.10	1.00	31.00
14	Shroud, Turbine	Inv Cst	310	1	7.64	18.00	0.12	1.20	19.20
15	Ring, Seal	Inv Cst	GM-60	2		0.40			0.40
16	Housing, Inlet	Snd Cst	GM-60	1	21.41	15.00	0.15	1.50	16.50
17	Screw, Machine	1/4-20	17-4	30	0.87	0.30	--	--	0.30
18	Nut	1/4-20	17-4	30	0.20	0.30	--	--	0.30
19	Housing, Diffuser	Inv Cst	GM-60	1	14.98	13.60	0.25	2.50	16.10
20	Support, Turbine	Inv Cst	Ni Rst	1	2.55	3.25	0.20	2.00	5.25
21	Ring, Seal	Inv Cst	Ni Rst	1		0.10	--	--	0.10
22	Ring, Pilot	Inv Cst	GM-60	1	0.35	0.50	0.025	0.25	0.75
23	Plate, Seal	Inv Cst	713C	1	0.49	4.50	0.065	0.65	5.15
24	Gasket	--	ASB	1	0.11	0.15	--	--	0.15

TABLE 8-3 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
25	Housing, Exhaust	Inv Cst	GM-60	1	27.92	21.00	0.22	2.20	23.20
26	Screw, Machine	1/4-20	Steel	36	1.16	0.36	--	--	0.36
27	Nut	Bar	Steel	36	0.23	0.36	--	--	0.36
28	Seal, Regenerator	Purch	Stl Ctd	1	0.50	5.00	--	--	5.00
29	Seal, Regenerator	Purch	Stl Ctd	1	0.50	5.00	--	--	5.00
30	Gasket	Stamp	Asb Cmp	1	0.80	0.20	--	--	0.20
31	Housing, Reg. Drive	Inv Cst	Dct Irn	1	5.03	3.60	0.085	0.85	4.45
32	Screw, Machine	1/4-1	Steel	16	0.46	0.12	--	--	0.12
33	Nut	Bar	Steel	16	0.10	0.16	--	--	0.16
34	Worm Gear, Regenerator	Bar	Brnze	1	0.25	3.64	--	--	3.64
35	Bearing, Worm Gear	Bar	Steel	1	0.05	0.06	0.025	0.25	0.31
36	Drive Motor, Regenerator			See Accessory List					
37	Gasket	Stamp	Asb Cmp	1	0.03	0.04	--	--	0.04
38	Cover	Inv Cst	Dct Irn	1	3.48	2.70	0.05	0.50	3.20
39	Seal	Stamp Mold	TFES+1	1	0.01	0.15	--	--	0.15
40	Bearing, Worm Wheel	PwrMet	Brnze	2	0.31	0.60	0.01	0.10	0.70
41	Wheel, Worm	Bar	4340	1	2.83	1.61	0.22	2.20	3.81
42	Housing, Center	SndCst	Ni Rst	1	60.98	52.00	0.28	2.80	54.80
43	Liner, Combustor	Sheet	1018	1	6.25	3.40	0.32	3.20	6.60
44	Cover	Snd Cst	GM-60	1	26.35	16.80	0.10	1.00	17.80
45	Clamp Cover	Purch	300S/S	1	--	1.50	--	--	1.50
46	Fitting, Fuel	Bar	300S/S	2	0.10	0.02	0.01	0.10	0.12
47	Gasket	Stamp	Asb Cmp	2	--	0.10	--	--	0.10
48	Tube Assembly, Fuel	Purch	Teflon	1	--	0.15	--	--	0.15

TABLE 8-3 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
49	Plenum Half, Upper	Snd Cst	GM-60	1	5.27	4.50	0.05	0.50	5.00
50	Plenum Half, Lower	Snd Cst	GM-60	1	2.77	2.75	0.05	0.50	3.25
51	Seal, Inlet	Mold	Plastc	1	0.10	0.05	--	--	0.05
52	Seal, Inlet	Mold	Plastc	1	0.10	0.05	--	--	0.05
53	Filter, Air			See Accessory List					
54	Cover, Filter	Mold	Plastc	1	1.54	0.18	--	--	0.18
55	Support, Actuator	Inv Cst	410S/S	1	0.47	2.50	0.035	0.35	2.85
56	Ring, Actuator	Inv Cst	440	1	0.78	6.50	0.055	0.55	7.05
57	Link, Actuator	Inv Cst	17-4	12	0.08	2.40	0.24	2.40	4.80
58	Pin, Actuator	Purch	410	12		0.06	--	--	0.06
59	Nut, Self-Locking	Bar	Steel	12	0.05	0.12	--	--	0.12
60	Bearing, Vane	Mold	Plastc	12	0.03	0.48	--	--	0.48
61	Vane Inlet	Inv Cst	410	12	0.27	6.00	0.24	2.40	8.40
62	Ring, Vane Support	Snd Cst	GM-60	1	1.92	2.00	0.15	1.50	3.50
	Ignition Plug	Purch	--	1	0.15	0.20	--	--	0.20
	Cover, Quill Drive	Tubing	AC	1	0.10	0.03	0.007	0.07	0.10
	H2O Injection Tubes	0.125D	S/S	3	0.15	0.90	0.01	0.10	1.00
	Acoustic Inlet Duct	Purch	Plastc	1	2.50	3.00	--	--	3.00
	Bolts Nuts Misc. Hardware and Unidentified Detail Parts				26.00	41.40	--	--	41.40
	Assembly Time					--	0.34	3.40	3.40
	Totals				256.35	380.56	4.815	48.15	428.71



SINGLE-SHAFT ENGINE WITH CERAMIC COUNTERFLOW RECUPERATOR,
VARIABLE INLET GUIDE VANES

FIGURE 8-6

TABLE 8-4
SINGLE-SHAFT ENGINE WITH CERAMIC COUNTERFLOW RECUPERATOR

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
1	Impeller	Inv Cst	17-4	1	1.80	18.00	0.38	3.80	21.80
2	Wheel, Turbine	Inv Cst	713C	1	4.11	35.00	0.52	5.20	40.20
3	Shaft, Wheel	Bar	4340	1	0.18	0.34	0.014	0.14	0.48
4	Shaft, Bearing	Forgng	4340	1	1.17	4.00	0.18	1.80	5.80
5	Foil Assembly, Thrust	Sheet	Inc750	2	0.03	0.30	0.04	0.40	0.70
6	Foil Assembly, Bearing	Sheet	Inc750	16	0.03	0.30	0.16	1.60	1.90
7	Carrier, Bearing	PwrMet	303	2	0.57	2.40	0.06	0.60	3.00
8	Coupling, Spline	Bar	4340	1	0.08	0.10	0.062	0.62	0.72
9	Nut	Bar	4340	1	0.02	0.03	0.022	0.22	0.25
11	Ring, Nozzle	Inv Cst	713C	1	1.23	20.00	0.12	1.20	21.20
12	Ring, Clamp	RldRng	310	1	0.94	7.00	0.08	0.80	7.80
13	Scroll, Turbine	Snd Cst	310	1	8.17	30.00	0.10	1.00	31.00
14	Shroud, Turbine	Inv Cst	310	1	7.64	18.00	0.12	1.20	19.20
15	Ring, Seal	Inv Cst	GM-60	2	--	0.40	--	--	0.40
16	Housing, Inlet	Snd Cst	GM-60	1	21.41	15.00	0.15	1.50	16.50
17	Screw, Machine	1/4-20	17-4	30	0.87	0.30	--	--	0.30
18	Nut	1/4-20	17-4	30	0.20	0.30	--	--	0.30
19	Housing, Diffuser	Inv Cst	GM-60	1	14.98	13.60	0.25	2.50	16.10
20	Support, Turbine	Inv Cst	Ni Rst	1	2.55	3.25	0.20	2.00	5.25
21	Ring, Seal	Inv Cst	Ni Rst	1	--	0.10	--	--	0.10
22	Ring, Pilot	Inv Cst	GM-60	1	0.35	0.50	0.025	0.25	0.75
23	Plate, Seal	Inv Cst	713C	1	0.49	4.50	0.065	0.65	5.15
24	Gasket	Purch	ASB	1	0.11	0.15	--	--	0.15

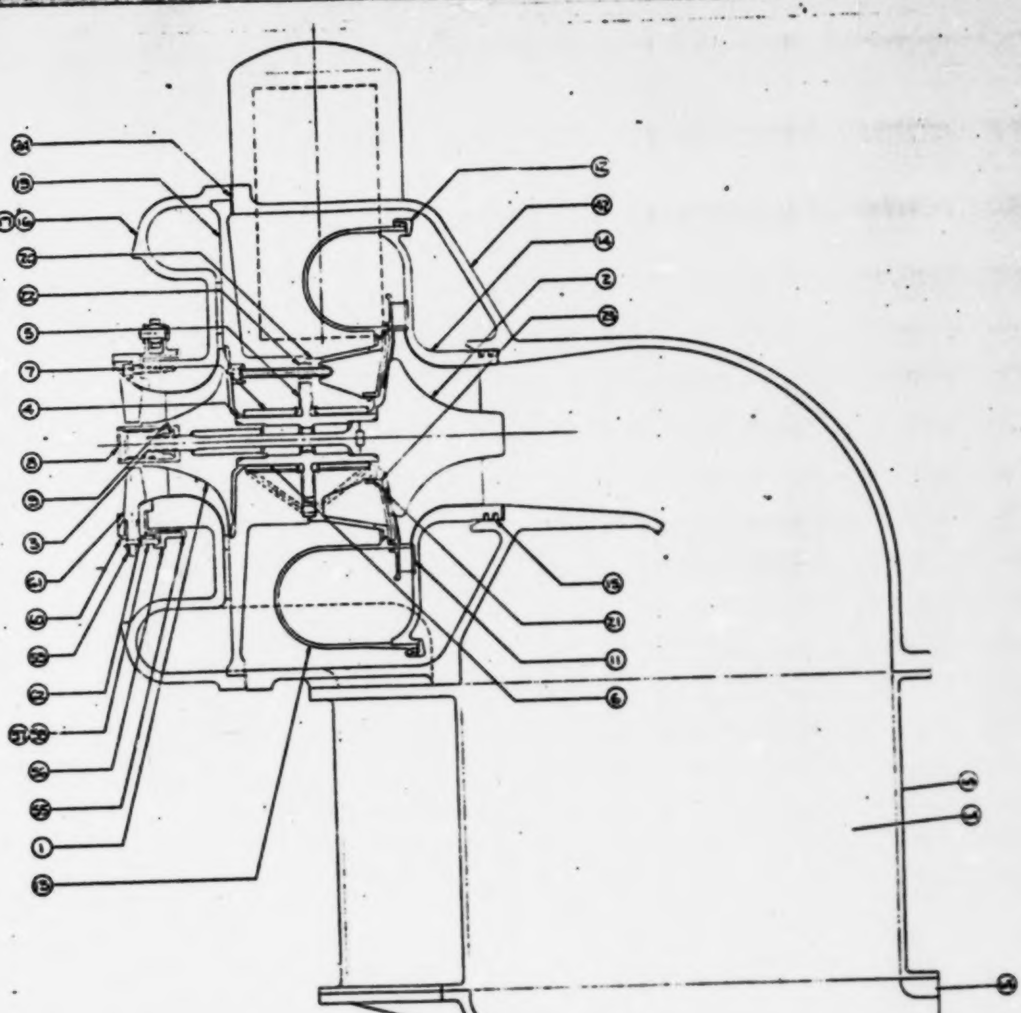
NOTE Item 10 - Not Assigned
Items 25 - 41 Not Assigned

TABLE 8-4 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
				36					
42	Housing, Center	Snd Cst	Ni Rst	1	60.98	52.00	0.28	2.80	54.80
43	Liner, Combustor	Sheet	1018	1	6.25	3.40	0.32	3.20	6.60
44	Cover	Snd Cst	GM-60	1	26.35	16.80	0.10	1.00	17.80
45	Clamp, Cover	Purch	300S/S	1	--	1.50	--	--	1.50
46	Fitting, Fuel	Purch	300S/S	2	0.10	0.02	0.01	0.10	0.12
47	Gasket	Purch	AsbCmp	2	--	0.10	--	--	0.10
48	Tube Assembly, Fuel	Purch	Teflon	1	--	0.15	--	--	0.15
49	Plenum Half, Upper	Snd Cst	GM-60	1	5.27	4.50	0.05	0.50	5.00
50	Plenum Half, Lower	Snd Cst	GM-60	1	2.77	2.75	0.05	0.50	3.25
51	Seal, Inlet	Mold	Plastc	1	0.10	0.05	--	--	0.05
52	Seal, Inlet	Mold	Plastc	1	0.10	0.05	--	--	0.05
53	Filter, Air			See Accessory List					
54	Cover, Filter	Mold	Plastc	1	1.54	0.18	--	--	0.18
55	Support, Actuator	Inv Cst	410S/S	1	0.47	2.50	0.035	0.35	2.85
56	Ring, Actuator	Inv Cst	440	1	0.78	6.50	0.055	0.55	7.05
57	Link, Actuator	Inv Cst	17-4	12	0.08	2.40	0.24	2.40	4.80
58	Pin, Actuator	Purch	410	12	--	0.06	--	--	0.06
59	Nut, Self-Locking	Purch	Steel	12	0.05	0.12	--	--	0.12
60	Bearing, Vane	Mold	Plastc	12	0.03	0.48	--	--	0.48
61	Bearing, Inlet	Inv Cst	410	12	0.27	6.00	0.24	2.40	8.40
62	Ring, Vane Support	Snd Cst	GM-60	1	1.92	2.00	0.15	1.50	3.50
63	Housing, Recuperator	Cstng	--	1	30.00	27.00	0.10	1.00	28.00
64	Recuperator	Fab	Ceramic	1	55.70	60.00	--	--	60.00

TABLE 8-4 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
65	Exhaust Manifold	Inv Cst	GM60	1	17.00	17.00	0.08	0.80	17.80
66	Seal, Recuperator	Purch	347S/S	1	0.20	1.25	--	--	1.25
67	Seal, Recuperator	Purch	347S/S	1	0.20	1.25	--	--	1.25
	Ignition Plug	Purch	--	1	0.15	0.20	--	--	0.20
	Cover, Quill Drive	Purch	AC	1	0.10	0.03	0.007	0.07	0.10
	H_2O Injection Tubes	0.125D	S/S	3	0.15	0.90	0.01	0.10	1.00
	Acoustic Inlet Duct	Mold	Plastc	1	2.50	3.00	--	--	3.00
	Bolts, Nuts, Misc Hardware and Unidentified Detail Parts				30.00	47.70	--	--	47.70
	Assembly Time				--	--	0.34	3.40	3.40
	TOTALS				309.99	433.46	4.615	46.15	479.61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SINGLE-SHAFT ENGINE WITH PLATE-FIN COUNTERFLOW RECUPERATOR VARIABLE INLET GUIDE VANES

FIGURE 8-7

TABLE 8-5
SINGLE-SHAFT ENGINE WITH PLATE-FIN COUNTERFLOW RECUPERATOR

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
1	Impeller	Inv Cst	17-4	1	1.80	18.00	0.38	3.80	21.80
2	Wheel, Turbine	Inv Cst	713C	1	4.11	35.00	0.52	5.20	40.20
3	Shaft, Wheel	Bar	4340	1	0.18	0.34	0.014	0.14	0.48
4	Shaft, Bearing	Forgng	4340	1	1.17	4.00	0.18	1.80	5.80
5	Foil Assembly, Thrust	Sheet	Inc750	2	0.03	0.30	0.04	0.40	0.70
6	Foil Assembly, Bearing	Sheet	Inc750	16	0.03	0.30	0.16	1.60	1.90
7	Carrier, Bearing	PwrMet	303	2	0.57	2.40	0.06	0.60	3.00
8	Spline, Coupling	Bar	4340	1	0.08	0.10	0.062	0.62	0.72
9	Nut	Bar	4340	1	0.02	0.03	0.022	0.22	0.25
11	Ring, Nozzle	Inv Cst	713C	1	1.23	20.00	0.12	1.20	21.20
12	Ring, Clamp	Rld Rng	310	1	0.94	7.00	0.08	0.80	7.80
13	Scroll, Turbine	Snd Cst	310	1	8.17	30.00	0.10	1.00	31.00
14	Shroud, Turbine	Inv Cst	310	1	7.64	18.00	0.12	1.20	19.20
15	Ring, Seal	Inv Cst	GM-60	2	--	0.40	--	--	0.40
16	Housing, Inlet	Snd Cst	GM-60	1	21.41	15.00	0.15	1.50	16.50
17	Screw, Machine	1/4-20	17-4	30	0.87	0.30	--	--	0.30
18	Nut	1/4-20	17-4	30	0.20	0.30	--	--	0.30
19	Housing, Diffuser	Inv Cst	GM-60	1	14.98	13.60	0.25	2.50	16.10
20	Support, Turbine	Inv Cst	Ni Rst	1	2.55	3.25	0.20	2.00	5.25
21	Ring, Seal	Inv Cst	Ni Rst	1		0.10	--	--	0.10
22	Ring, Pilot	Inv Cst	GM-60	1	0.35	0.50	0.025	0.25	0.75
23	Plate, Seal	Inv Cst	713C	1	0.49	4.50	0.065	0.65	5.15
24	Gasket	Purch	ASB	1	0.11	0.15	--	--	0.15

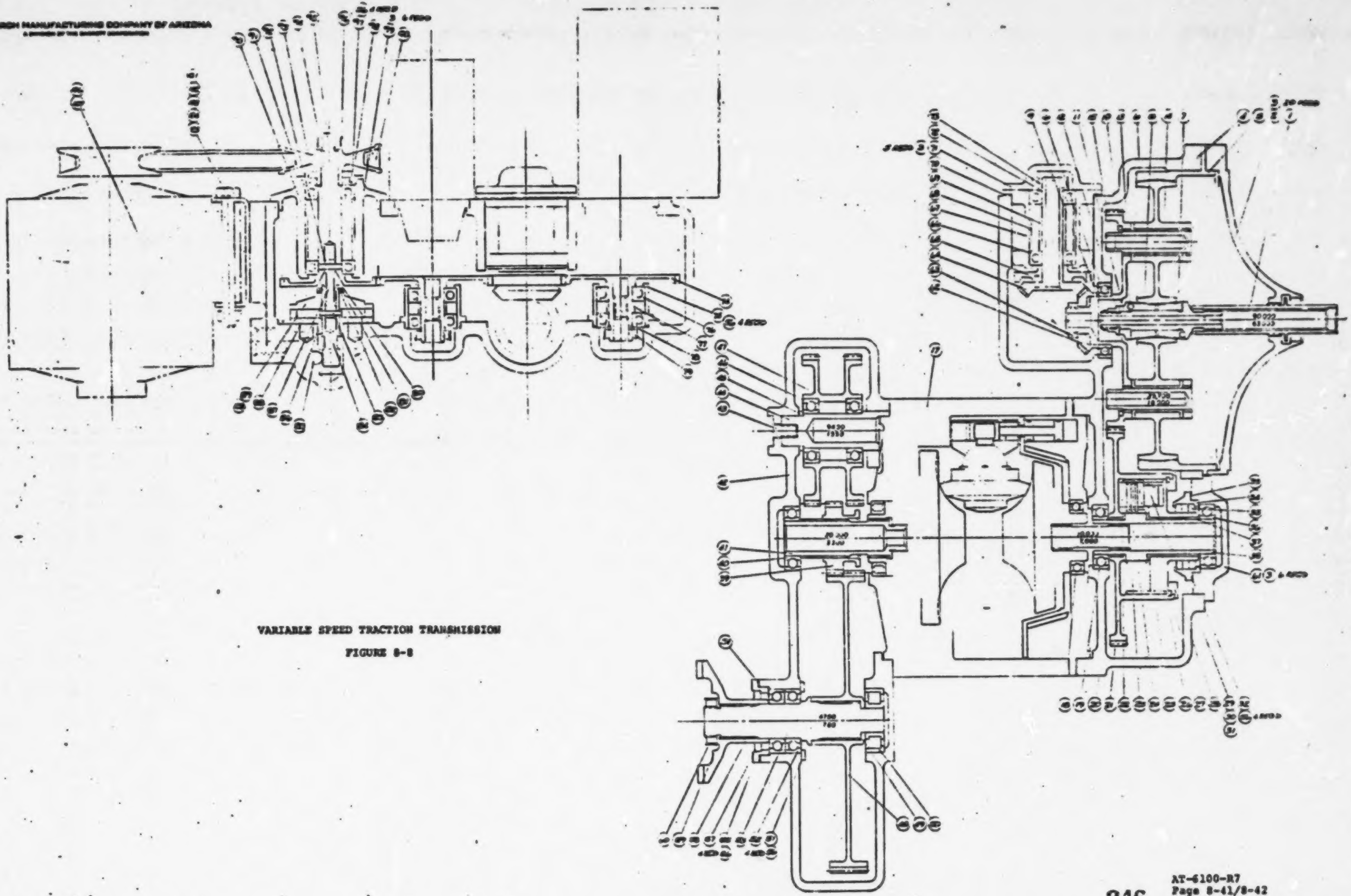
Item 10 - Not Assigned
Items 25 - 41 Not Assigned

TABLE 8-5 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
	Nut								
42	Housing, Center	Snd Cst	N_1 Rst	1	60.98	52.00	0.28	2.80	54.80
43	Liner, Combustor	Sheet	1018	1	6.25	3.40	0.32	3.20	6.60
44	Cover	Snd Cst	GM-60	1	26.35	16.80	0.10	1.00	17.80
45	Clamp, Cover	Purch	300S/S	1	--	1.50	--	--	1.50
46	Fitting, Fuel	Bar	300S/S	2	0.10	0.02	0.01	0.10	0.12
47	Gasket	Stamp	AbsCmp	2	--	0.10	--	--	0.10
48	Tube Assembly, Fuel	Purch	Teflon	1	--	0.15	--	--	0.15
49	Plenum Half, Upper	Snd Cst	GM-60	1	5.27	4.50	0.05	0.50	5.00
50	Plenum Half, Lower	Snd Cst	GM-60	1	2.77	2.75	0.05	0.50	3.25
51	Seal, Inlet	Mold	Plastc	1	0.10	0.05	--	--	0.05
52	Seal, Inlet	Mold	Plastc	1	0.10	0.05	--	--	0.05
53	Filter, Air			See Accessory List					
54	Cover, Filter	Mold	Plastc	1	1.54	0.18	--	--	0.18
55	Support, Actuator	Inv Cst	410S/S	1	0.47	2.50	0.035	0.35	2.85
56	Ring, Actuator	Inv Cst	440	1	0.78	6.50	0.055	0.55	7.05
57	Link, Actuator	Inv Cst	17-4	12	0.08	2.40	0.24	2.40	4.80
58	Pin, Actuator	Purch	410	12	--	0.06	--	--	0.06
59	Nut, Self-Locking	Bar	Steel	12	0.05	0.12	--	--	0.12
60	Bearing, Vane	Mold	Plastc	12	0.03	0.48	--	--	0.48
61	Vane, Inlet	Inv Cst	410	12	0.27	6.00	0.24	2.40	8.40
62	Ring, Vane Support	Snd Cst	GM-60	1	1.92	2.00	0.15	1.50	3.50
63	Housing, Recuperator	Cstng	GM-60	1	45.00	40.00	0.10	1.00	41.00
64	Recuperator	Fab	347	1	63.00	94.50	--	--	94.50

TABLE 8-5 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
65	Exhaust Manifold	Cstng	GM60	1	22.00	15.40	0.08	0.80	16.20
66	Seal, Recuperator	Purch	347S/S	1	0.20	1.25	--	--	1.25
67	Seal, Recuperator	Purch	347S/S	1	0.20	1.25	--	--	1.25
	Ignition Plug	Purch	--	1	0.15	0.20	--	--	0.20
	Cover, Quill Drive	Tubing	AC	1	0.10	0.03	0.007	0.07	0.10
	H_2O Injection Tubes	0.125D --	S/S	3	0.15	0.90	0.01	0.10	1.00
	Acoustic Inlet Duct	Purch	Plastc	1	2.50	3.00	--	--	3.00
	Miscellaneous Nuts, Bolts,								
	Unidentified Details				30.00	47.10			47.70
	Assembly Time						0.34	3.40	3.40
	TOTALS				337.29	479.36	4.615	46.15	525.51



VARIABLE SPEED TRACTION TRANSMISSION

FIGURE 8-8

TABLE 8-6
VARIABLE SPEED TRACTION TRANSMISSION

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
1	Shaft, Quill	Bar	4340	1	0.13	0.36	0.14	1.40	1.76
2	Housing, Inlet	Inv Cst	GM-60	1	5.05	3.45	0.10	1.00	4.45
3	Screw, Machine	Purch	LC Stl	20	0.01	0.10	--	--	0.10
4	Housing, Gear	Snd Cst	GM-60	1	17.06	13.50	0.26	2.60	16.10
5	Ring, Planetary	Forgng	4620	1	4.75	2.80	0.37	3.70	6.50
6	Planet	Forgng	4620	3	3.22	2.25	0.30	3.00	5.25
7	Sun Planetary	Bar	4620	1	0.13	0.04	0.08	0.80	0.84
8	Sun Planetary	Bar	4620	1	0.13	0.04	0.08	0.80	0.84
9	Shaft, Input	Bar	4620	1	0.25	0.07	0.12	1.20	1.27
10	Ball	Purch	4620	6	Inc.in 7,8,11	0.18	--	--	0.18
11	Spacer, Sun	Bar	4620	1	0.04	0.02	0.06	0.60	0.62
12	Nut	Bar	4340	1	0.03	0.015	0.025	0.25	0.26
13	Carrier, Planetary	Forgng	4620	1	1.79	1.18	0.28	2.80	3.98
14	Bearing, Planet	Pwr Mtl	Brnze	6	0.34	1.20	0.02	0.20	1.40
15	Shaft, Planet	Bar	4340	3	0.55	0.16	0.06	0.60	0.76
16	Bearing, Ball	Purch	52100	1	0.17	0.96	--	--	0.96
17	Transmission, Traction	Purch	Misc	1	35.00	15.00	6.0	60.0	75.00
18	Shaft Quill	Bar	4620	1	1.28	0.30	0.082	0.82	1.12
19	Bearing, Ball	Purch	52100	1	0.15	0.96	--	--	0.96
20	Bearing, Sleeve	Pwr Met	Brnze	1	0.04	0.15	0.01	0.10	0.25
21	Gear, Clutch	Forgng	4620	1	2.30	1.50	0.58	5.80	7.30
22	Spacer, Disc	Pwr Mtl	4340	1		--	--	--	
23	Disc, Friction	Stamp	Cu-Grap	3		--	--	--	
24	Disc, Friction	Stamp	4340	3	1.79	--	--	--	

TABLE 8-6 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
25	Spacer, Disc	Pwr Met	4340	1	--	--	--	--	
26	Cover, Clutch	Pwr Met	CstIrn	1	0.66	4.80	--	--	4.80
27	Ring, Retaining	Purch	SprStl	1	Inc.in 21, 28	--	--	--	
28	Plate, Clutch	Pwr Met	4340	1	0.27	--	--	--	
29	Piston, Clutch	Pwr Met	GhtIrn	1	0.53	--	--	--	
30	O-Ring	Mold	Silcon	1	--	0.02	--	--	0.02
31	O-Ring	Mold	Silcon	1	--	0.02	--	--	0.02
32	Carrier Seal	Pwr Met	FM Irn	1	0.22	0.30	0.025	0.25	0.55
33	Gasket	Stamp	Cmpsit	1	--	0.02	--	--	0.02
34	Spacer, Bearing	Pwr Met	FM Irn	1	0.04	0.15	--	--	0.15
35	Seal, Lip	Stamp	TFE	2	0.02	0.30	--	--	0.30
36	Bearing, Ball	Purch	52100	1	0.17	0.96	--	--	0.96
37	Bearing, Sleeve	Tubing	Brnze	1	0.11	0.88	0.026	0.26	1.14
38	Shaft, Clutch	Bar	4130	1	0.51	0.20	0.18	1.80	2.00
39	Gear, Spur	Pwr Met	4620	1	0.39	1.80	0.025	0.25	2.05
40	Bearing, Ball	Purch	52100	1	0.17	0.76	--	--	0.76
41	Shaft, Spline	Bar	4130	1	0.42	0.20	0.11	1.10	1.30
42	Housing, Gear	Snd Cst	GM-60	1	17.52	12.00	0.26	2.60	14.60
43	Screw, Machine	Purch	FM	1	0.01	0.03			0.03
44	Shaft, Idler	Bar	1117	1	0.47	0.18	0.05	0.50	0.68
45	Spacer, Idler	Pwr Met	PwrIrn	1	0.02	0.15	--	--	0.15
46	Bearing, Ball	Purch	52100	2	0.34	1.92	--	--	1.92
47	Gear, Idler	Pwr Met	4620	1	3.93	4.50	0.17	1.70	6.20
48	Gear, Output	Forgng	4620	1	4.10	2.50	0.23	2.30	4.80

TABLE 8-6 (Contd)

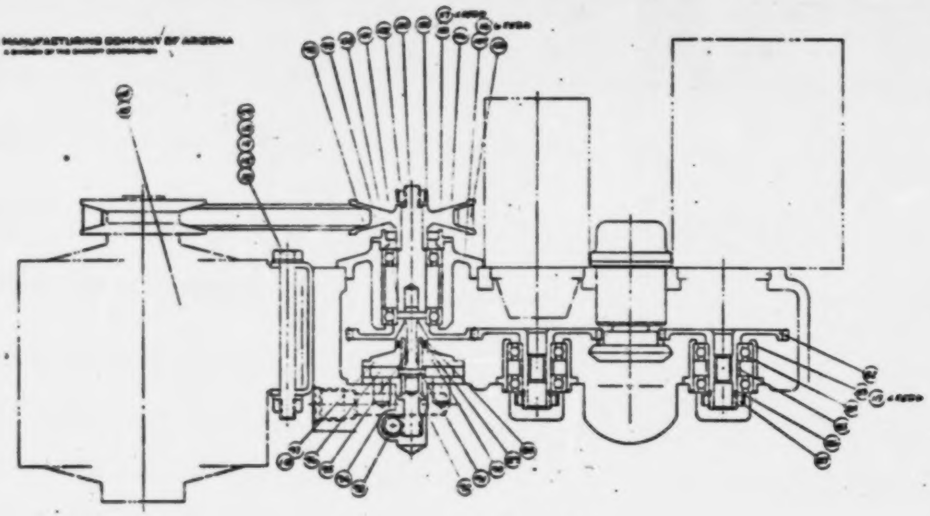
FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
49	Spacer	Pwr Met	Iron	1	0.05	0.25	0.01	0.10	0.35
50	Bearing, Roller	Purch	52100	1	0.28	1.25	--	--	1.25
51	Retainer, Bearing	Pwr Met	Iron	1	0.05	0.20	--	--	0.20
52	Bearing, Ball	Purch	52100	2	0.34	1.92	--	--	1.92
53	Shaft, Output	Bar	4130	1	1.12	0.54	0.15	1.50	2.04
54	Gasket	Stamp	Cmpsit	1	--	0.03	--	--	0.03
55	Retainer, Seal	Pwr Met	Iron	1	0.18	0.25	0.015	0.15	0.40
56	Screw, Machine	Purch	Steel	4	0.04	0.12			0.12
57	Seal, Lip	Stamp	TFE	1	0.01	0.15	--	--	0.15
58	Flange, Output	Forgng	6260	1	1.00	0.70	0.10	1.00	1.70
59	Washer, Lock	Stamp	1018	1	0.04	0.01	0.005	0.05	0.06
60	Nut	Bar	1113	1	0.06	0.10	--	--	0.10
61	Bearing, Cover	Inv Cst	GM-60	1	1.37	1.00	0.05	0.50	1.50
62	Gear, Bevel Spiral	Bar	6260	1	0.20	0.04	0.50	5.00	5.04
63	Nut	Bar	6260	1	0.02	0.02	0.01	0.10	0.12
64	Gear, Bevel Spiral	Forgng	4620	1	0.43	0.39	0.58	5.80	6.19
65	Gear, Spur	Bar	4620	1	0.12	0.03	0.055	0.55	0.58
66	Bearing, Ball	Purch	52100	2	0.30	1.06	--	--	1.06
67	Carrier, Bearing	Cstng	4620	1	0.40	0.08	0.08	0.80	0.88
68	Spacer, Bearing	Tubing	4620	1	0.11	0.02	0.01	0.10	0.12
69	Spacer, Bearing	Tubing	4620	1	0.02	0.02	0.01	0.10	0.12
70	Shim	Stamp	Steel	A/R	A/R	0.03	--	--	0.03
71	Cover	Cstng	GM-60	1	0.39	0.50	0.015	0.15	0.65
72	Ring, Lock	Stamp	MilStl	1		0.01	--	--	0.01

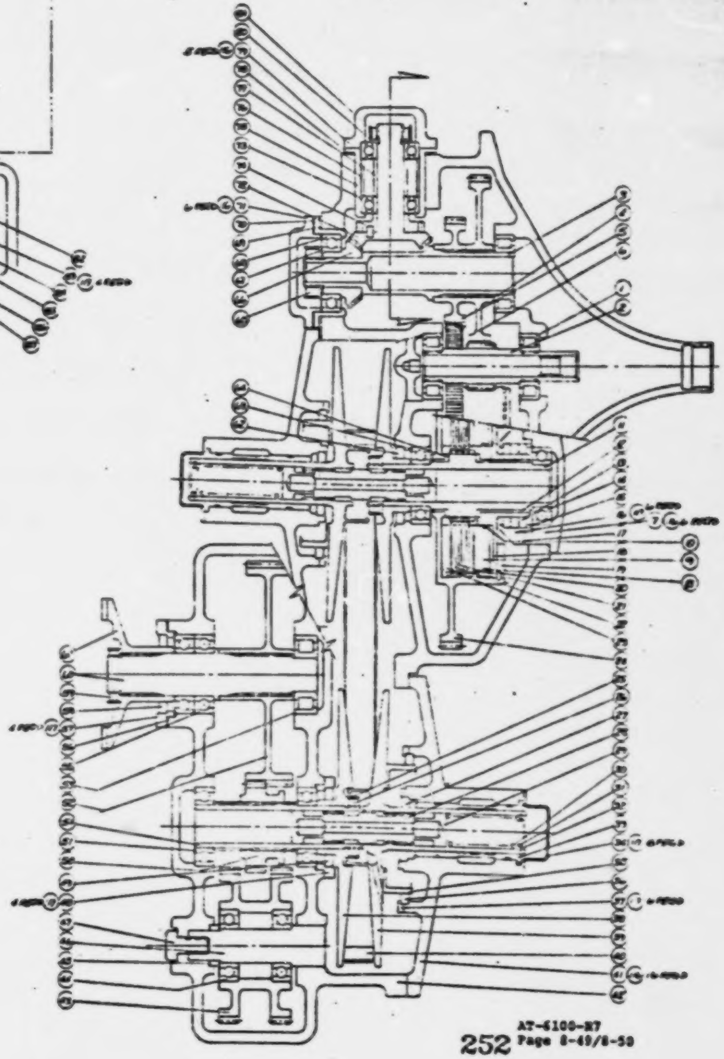
TABLE 8-6 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
73	Nut	Bar	4620	1	0.05	0.01	0.015	0.15	0.16
74	Gear, Spur	Forgng	4620	2	1.32	0.89	0.08	0.88	1.77
75	Carrier, Bearing	Tubing	4620	2	0.52	0.10	0.05	0.50	0.60
76	Bearing, Ball	Purch	52100	4	0.60	2.12	--	--	2.12
77	Spacer, Bearing	Tubing	4620	2	0.02	0.01	0.003	0.03	0.04
78	Ring, Lock	Stamp	MilSTL	2	0.10	0.01	--	--	0.01
79	Nut	Bar	4620	1	0.03	0.01	0.015	0.15	0.16
80	Ring, Retaining	Purch	Steel	1	0.01	0.03	--	--	0.03
81	Ring, Retaining	Purch	Steel	1	See	0.01	--	--	0.01
82	Washer, Thrust	Stamp	Steel	1	Item	0.04	--	--	0.04
83	Pin	Purch	Steel	1	89	0.01	--	--	0.01
84	Gear, Worm	Bar	4620	1		0.04	0.096	0.96	1.00
85	Drive, Tachometer	Bar	4620	1	0.11	0.02	0.09	0.90	0.92
86	Housing, Lube Pump	Pwr M+1	GM-60	1	Item 89	1.00	0.04	0.40	1.40
87	Shaft	Bar	4340	1	0.02	0.02	0.02	0.20	0.22
88	Rotor Set, Lube Pump	Pwr M+1	Steel	1	Item 89	0.75	--	--	0.75
89	Housing, Lube Pump	Pwr M+1	GM-60	1	0.82	0.60	0.005	0.05	0.65
90	Spacer, Bearing	Tubing	4620	1	0.10	0.02	0.01	0.10	0.12
91	Bearing, Ball	Purch	52100	2	0.20	1.06	--	--	1.06
92	Pin	Purch	Steel	1	--	0.01	--	--	0.01
93	Seal, Lip	Stamp	TFES	1	0.01	0.15	--	--	0.15
94	Nut	Bar	4620	1	0.02	0.01	0.015	0.15	0.16
95	Gear, Spur	Forgng	4620	1	0.58	0.50	0.12	1.20	1.70
96	Ring, Lock	Stamp	MilSTL	1	0.01	0.01	--	--	0.01

TABLE 8-6 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
97	Retainer, Bearing	Tubing	4130	1	0.40	0.20	0.015	0.15	0.35
98	Sheave, V-Belt	Cstng	Iron	1	0.44	0.65	--	--	0.65
99	Carrier, Bearing	Inv Cst	GM-60	1	1.64	0.09	0.08	0.80	0.89
100	V-Belt	Mold	Rubber	1	0.23	0.60	--	--	0.60
101	Bolt	Bar	Steel	1	0.03	0.07	--	--	0.07
102	Washer	Stamp	Steel	1	0.18	0.01	--	--	0.01
103	Washer, Lock	Stamp	Steel	1	0.01	0.01	--	--	0.01
104	Nut	Bar	Steel	1	0.02	0.03	--	--	0.03
105	Bracket	Stamp	Steel	1	0.17	0.05	0.005	0.05	0.10
106	Alternator				See Accessory List				
107	Comp, Air Conditioner				See Accessory List				
108	Screw, Machine	Bar	Steel	6	0.12	0.06	--	--	0.06
	Bolts, Nuts Misc Hardware and								
	Unidentified Detail Parts				10.50	12.36	--	--	12.36
	Assembly Time					--	0.50	5.00	5.00
	TOTALS				128.91	106.20	12.400	124.00	230.20





VARIABLE SPEED BELT TRANSMISSION

FIGURE 8-9

TABLE 8-7
VARIABLE SPEED BELT TRANSMISSION

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
1	Bearing, Roller	Purch	M-50	2	0.33	16.00	--	--	16.00
2	Gear, Pinion	Bar	6260	1	0.31	0.05	0.16	1.60	1.65
3	Shaft, Idler	Bar	6260	1	0.50	0.68	0.08	0.80	1.48
4	Bearing, Roller	Purch	52100	1	0.20	1.20	--	--	1.20
5	Gear, Spur	Pwr Mtl	6260	1	0.50	1.50	0.025	0.25	1.75
6	Gear, Spur	Bar	6260	1	1.21	0.55	0.19	1.90	2.45
7	Cover	Inv Cst	M-60	1	2.88	1.75	0.05	0.50	2.25
8	O-Ring	Mold	VitonA	1	--		See Page 5		
9	Piston, Clutch	Pwr Mtl	Grftzd	1	0.74	Included in Items 18 - 23			
10	O-Ring	Mold	VitonA	1	--		See Page 5		
11	Shaft, Clutch	Bar	6260	1	0.67	0.20	0.12	1.20	1.40
12	Bearing, Sleeve	Tubing	Brnze	1	0.13	0.08	0.026	0.26	0.34
13	Seal, Lip	Stamp	TFESTL	2	0.02	0.30	--	--	0.30
14	Bearing, Ball	Purch	52100	1	0.16	0.96	--	--	0.96
15	Spacer	Pwr Mtl	FM STL	1	0.02	0.15	--	--	0.15
16	Carrier, Seal	Pwr Mtl	FM STL	1	0.39	0.30	0.025	0.25	0.55
17	Gasket	Stamp	Compst	1	--	0.02	--	--	0.02
18	Carrier, Piston	Pwr Mtl	CstIrn	1	0.83	--	--	--	
19	Plate, Retaining	Pwr Mtl	Steel	1	0.61	--	--	--	
20	Ring, Retaining	Stamp	SprStl	1	Inc.in 19, 24	4.80	--	--	4.80
21	Disc, Friction	Stamp	Cu-Grp	4	1.82	--	--	--	
22	Disc, Friction	Stamp	4340	4	--	--	--	--	
23	Spacer, Disc	Pwr Mtl	4340	1	0.68	--	--	--	
24	Gear, Spur	Forgng	6260	1	3.04	1.90	0.58	5.80	7.70

TABLE 8-7 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
25	Ring, Lock	Stamp	1018	2	0.06	0.01	0.005	0.05	0.06
26	Nut	Tubing	4130	2	0.10	0.02	0.015	0.15	0.17
27	Seal, Lip	Stamp	TFEStl	2	0.02	0.30	--	--	0.30
28	Shaft, Quill	Bar	4140	2	0.40	0.10	0.12	1.20	1.30
29	Plunger	Pwr Mtl	4600	2	0.05	0.30	0.02	0.20	0.50
30	Spring	Wire	CrbStl	2		0.06	--	--	0.06
31	Retainer	Stamp	MilStl	2	0.04	0.06	--	--	0.06
32	Ring, Retainer	Stamp	SprStl	2		0.02	--	--	0.02
33	Bearing, Sleeve	Tubing	Brnze	2	0.72	0.57	0.05	0.50	1.07
34	Cover	Stamp	MilStl	2	0.14	0.02	0.01	0.10	0.12
35	Piston, Sheave	Bar	4130	2	0.59	0.70	0.04	0.40	1.10
36	Seal, Lip	Stamp	TFEStl	2	0.05	0.30	--	--	0.30
37	Retainer	Bar	LB Stl	2	0.22	0.30	--	--	0.30
38	Sheave, Stationary	Bar Cst	Stl w/ Brnze	2	4.32	1.30	--	--	1.30
39	Sheave, Movable	Bar Cst	Stl w/ Brnze	2	5.09	1.92	--	--	1.92
40	Belt	Mold	Cmpsit	1	0.35	2.50	--	--	2.50
41	Housing, Sheave Support	Inv Cst	GM-60	2	9.20	7.00	0.11	1.10	8.10
42	Housing, Transmission	Snd Cst	GM-60	1	55.91	44.80	0.45	4.50	49.30
43	Gear, Idler	Pwr Mtl	4600	1	1.84	3.00	0.14	1.40	4.40
44	Bearing, Ball	Purch	52100	2	0.30	1.92	--	--	1.92
45	Spacer	Pwr Mtl	4600	1	0.05	0.05	--	--	0.05
46	Shaft, Idler	Bar	4130	1	0.59	0.08	0.02	0.20	0.28
47	Screw, Machine	3/8x1.0	FM Stl	1		0.02	--	--	0.02
48	Retainer, Seal	Tubing	4130	1	0.16	0.02	0.02	0.20	0.22

TABLE 8-7 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
49	Seal, Lip	Stamp	TFE/Stl	1	0.01	0.15	--	--	0.15
50	Gear, Spur	Pwr Mtl	4600	1	0.46	1.80	0.025	0.25	2.05
51	Bearing, Ball	Purch	52100	2	0.34	1.42	--	--	1.42
52	Gear, Output	Forgng	4620	1	2.15	1.38	0.10	1.00	2.38
53	Bearing, Roller	Purch	52100	1	0.28	1.25	--	--	1.25
54	Shaft, Sheave	Bar	4340	1	0.45	0.10	0.09	0.90	1.00
55	Bearing, Ball	Purch	52100	2	0.35	1.92	--	--	1.92
56	Gasket	Sheet	Cmpsit	1	--	0.01	--	--	0.01
57	Retainer, Seal	Tubing	4130	1	0.20	0.03	0.02	0.20	0.23
58	Seal, Lip	Stamp	TFE/Stl	1	0.01	0.15	--	--	0.15
59	Nut	Bar	4620	1	0.06	0.01	0.015	0.15	0.16
60	Shaft, Output	Bar	4130	1	1.06	0.16	0.08	0.80	0.96
61	Flange, Output	Forgng	6260	1	0.77	0.70	0.10	1.00	1.70
62	Seal, Lip	Stamp	TFE/Stl	1	0.01	0.15	--	--	0.15
63	Bearing, Ball	Purch	52100	1	0.16	0.96	--	--	0.96
64	Bearing, Sleeve	Stamp	Brnze	1	0.05	0.15	0.01	0.10	0.25
65	Nut	Bar	FM Stl	1	0.09	0.02	0.015	0.15	0.17
66	Gear, Bevel	Bar	6260	1	0.20	0.04	0.50	5.00	5.04
67	Ring, Lock	Stamp	MilStl	1	See 65	0.01	0.005	0.05	0.06
68	Bearing, Ball	Purch	52100	1	0.16	0.76	--	--	0.76
69	Shim	Sheet	S/S	A/R	See 42	0.01	--	--	0.01
70	Gasket	Sheet	Cmpsit	1	--	0.02	--	--	0.02
71	Cover	Inv Cst	GM-60	1	0.42	0.75	0.02	0.20	0.95
72	Gear, Bevel	Forgng	4620	1	0.43	0.39	0.58	5.80	6.19

TABLE 8-7 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
73	Bearing, Ball	--	52100	2	0.19	1.06	--	--	1.06
74	Shaft, Accessory Drive	Bar	4620	1	0.12	0.03	0.05	0.50	0.53
75	Carrier, Bearing	Pwr Mtl	4600	1	0.40	0.75	0.01	0.10	0.85
76	Spacer, Bearing, Outer	Tubing	Steel	1	0.11	0.02	0.015	0.15	0.17
77	Spacer, Bearing, Inner	Tubing	Steel	1	--	0.02	0.015	0.15	0.17
78	Shim	Stamp	Cmpsit	A/R	See 75	0.01	--	--	0.01
79	Cover	Inv Cst	GM-60	1	0.39	0.50	0.025	0.25	0.75
80	Ring, Lock	Stamp	1018	1	0.05	0.01	0.005	0.05	0.06
81	Nut	Bar	4620	1	--	0.01	0.015	0.15	0.16
82	Gear, Spur	Forgng	4130	2	1.32	0.95	0.30	3.00	3.95
83	Carrier, Bearing	Pwr Mtl	4600	2	0.52	1.20	0.01	0.10	1.30
84	Bearing, Ball	Purch	52100	4	0.41	2.12	--	--	2.12
85	Spacer, Bearing	Tubing	Al	2	See 82	0.01	0.003	0.03	0.04
86	Ring, Lock	Stamp	MilStl	2	0.10	0.02	0.01	0.10	0.12
87	Nut	Bar	4130	2	--	0.02	0.03	0.30	0.32
88	Ring, Retaining	Stamp	MilStl	1	See 97	0.03	--	--	0.03
89	Ring, Retaining	Stamp	MilStl	1	--	0.01	--	--	0.01
90	Washer, Thrust	Bar	Steel	1	See 97	0.04	--	--	0.04
91	Pin	Purch	Steel	1	--	0.01	--	--	0.01
92	Shaft, Gear, Worm	Bar	4620	1	0.10	0.02	0.088	0.88	0.90
93	Drive, Tachometer	Bar	4620	1	0.11	0.02	0.09	0.90	0.92
94	Housing, Pump, Lube	Pwr Mtl	GM-60	1	See 97	1.00	0.04	0.40	1.40
95	Shaft, Quill	Bar	4340	1	0.02	0.02	0.02	0.20	0.22
96	Rotor Set, Pump, Lube	Pwr Mtl	Steel	1	See 97	0.75	--	--	0.75

TABLE 8-7 (Contd)

FIND NO.	DESCRIPTION	PROCESS	MATERIAL	QUANTITY	FINISHED WEIGHT	MATERIAL COST	MANUFACTURING - HOURS	MANUFACTURING COST - DOLLARS	COST - TOTAL
97	Housing, Pump, Lube	PwrMtl	GM-60	1	0.82	0.60	0.005	0.05	0.65
98	Spacer, Bearing	Tubing	4620	1	0.10	0.02	0.01	0.10	0.12
99	Bearing, Ball	Purch	52100	2	0.20	1.06	--	--	1.06
100	Pin	Purch	Steel	1	See 95	0.01	--	--	0.01
101	Seal, Lip	Stamp	TFE/Stl	1	0.01	0.15	--	--	0.15
102	Nut	Bar	4130	1		0.01	0.015	0.15	0.16
103	Gear, Spur	Forgng	4620	1	0.58	0.50	0.120	1.20	1.70
104	Ring, Lock	Stamp	TFE/Stl	1		0.01	--	--	0.01
105	Retainer, Bearing	Tubing	4130	1	0.10	0.20	0.015	0.15	0.35
106	Sheave, U-Belt	Cstng	Iron	1	0.44	0.65	--	--	0.65
107	Housing, Bearing	InvCst	M-60	1	1.64	1.05	0.05	0.50	1.55
108	V-Belt	Mold	Rubber	1	0.23	1.00	--	--	1.00
109	Bolt	Bar	Steel	1		0.07	--	--	0.07
110	Nut	Bar	Steel	1	0.18	0.03	--	--	0.03
111	Washer	Stamp	Steel	1		0.01	--	--	0.01
112	Washer, Lock	Stamp	Steel	1		0.01	--	--	0.01
113	Bracket	Stamp	Steel	1	0.17	0.05	0.005	0.05	0.10
114	Alternator			See Accessory List					
115	Comp, Air Conditioner			See Accessory List					
116	Bolt	Bar	Steel	39	1.17	0.78	--	--	0.78
117	Screw	Bar	Steel	36	0.72	0.36	--	--	0.36
118	Screw	Bar	Steel	6	0.12	0.06	--	--	0.06
	Bolts, Nuts Misc Hrd & Unident.	Detail Parts		--	11.25	15.64	---	15.64	15.64
	Assembly Time						0.50	5.00	5.00
TOTALS					124.47	139.05	5.262	52.62	191.67

TABLE 8-8

AUTOMOBILE GAS TURBINE SYSTEM CONTROLS COST AND WEIGHT

Systems	Engine Type			
	Single-Shaft		Free-Turbine	
	Factory Cost, Dollars	Weight, lb	Factory Cost, Dollars	Weight, lb
FUEL CONTROL				
Hydromechanical				
Fuel Filter	1.07	0.2	1.07	0.2
Fuel Pump	6.81	2.0	6.81	2.0
Relief Valve and Spring	.50	0.1	.50	0.1
Metering Section	6.47	1.0	6.47	1.0
Governor	11.90	2.0	14.40	2.3
By-Pass Valve	5.43	0.8	5.43	0.8
Fuel Solenoid	6.65	0.5	6.65	0.5
Governor Reset			10.00	1.0
TOTAL	38.83	6.6	51.33	7.9
Electronic				
Fuel Filter	1.07	0.2	1.07	0.2
Fuel Pump	6.81	2.0	6.81	2.0
Relief Valve and Spring	.50	0.1	.50	0.1
Metering Section	13.30	0.8	13.30	0.8
By-Pass Valve	5.43	0.8	5.43	0.8
Fuel Solenoid	6.65	0.5	6.65	0.5
Computing Section	15.60	0.6	17.53	0.7
TOTAL	49.36	5.0	51.29	5.1
ACTUATOR				
Hydromechanical				
Power Pistor and Rod	7.50	2.5	15.00	5.0
Servo Spool	4.43	0.5	8.86	1.0
Brake Servo	2.50	0.5	5.00	1.0
Housing	5.00	4.5	10.00	9.0
Linkage	1.50	0.5	3.00	1.0
TOTAL	20.93	8.5	41.86	17.0

TABLE 8-8 (CONTD)

Systems	Engine Type			
	Single-Shaft		Free-Turbine	
	Factory Cost, Dollars	Weight, lb	Factory Cost, Dollars	Weight, lb
WATER INJECTION CONTROL				
Temperature Sensor	7.50	0.3	N/A	N/A
Needle Valve	1.50	0.1		
TOTAL	9.00	0.4		
SENSORS				
Hydromechanical				
Overtemperature	8.00	0.4	Same As Single-Shaft	
Overspeed	5.00	0.3		
TOTAL	13.00	0.7		
Electronic				
Temperature (Thermocouples)	10.00	0.1	Same As Single-Shaft	
Speed	5.00	0.3		
TOTAL	15.00	0.4		

TABLE 8-8 (CONTD)

Systems	Engine Type			
	Single-Shaft		Free-Turbine	
	Manufacturing Cost, Dollars	Weight, lb	Manufacturing Cost, Dollars	Weight, lb
ACTUATOR				
Electronic				
Power Piston and Rod	7.50	2.5	15.00	5.0
Servo Spool	4.43	0.5	8.86	1.0
Housing	5.00	4.5	10.00	9.0
Driver	13.30	0.8	26.60	1.6
Computing Section	7.72	0.3	15.44	0.6
TOTAL	37.95	6.6	75.90	13.2
AUTOMATIC STARTING AND PROTECTION CONTROL				
Hydromechanical System	23.41	0.9	25.16	1.0
Electronic System	15.44	0.6	17.34	0.7
TRANSMISSION CONTROL				
Speed Servo and Housing	12.20	1.6	15.00	2.0
Feed Back Linkage	4.00	.2	N/A	N/A
HYDRAULIC CLUTCH CONTROL				
Servo Spool and Housing	3.35	1.0	3.5	1.0
Feedback Linkage	1.50	0.1	N/A	N/A
Engine Speed Servo	11.00	1.5	11.00	1.5
TOTAL	15.85	2.6	14.50	2.5

TABLE 8-9

CONTROL SYSTEM COST AND WEIGHT SUMMARY

Systems	Single-Shaft Engine				Free-Turbine Engine			
	Hydromechanical		Electronic		Hydromechanical		Electronic	
	Factory Cost, Dollars	Weight, lb	Factory Cost, Dollars	Weight, lb	Factory Cost, Dollars	Weight, lb	Factory Cost, Dollars	Weight, lb
Fuel Control	38.83	6.6	49.36	5.0	51.33	7.9	51.29	5.1
Actuator	20.93	8.5	37.95	6.6	41.86	17	75.90	13.2
Automatic Starting and Protection Control	23.41	0.9	15.44	0.6	25.16	1.0	17.34	0.7
Transmission Control	16.20	1.8	16.20	1.8	15.00	2.0	15.00	2.0
Hydraulic Clutch Control	15.85	2.6	15.85	2.6	14.50	2.5	14.50	2.5
Maximum Power Control	9.00	0.4	9.00	0.4	--	--	--	--
Sensors	13.00	0.7	15.00	0.4	13.00	0.7	15.00	0.9
Total	137.22	21.5	158.80	17.4	160.85	31.1	199.03	24.9

These estimates are based on the system schematics and the discussion in the supplement of this report. Table 8-10 is a summary of accessories weights and costs, and Tables 8-11 through 8-14 summarize weights and costs for the various engine-controls-transmission systems. A summary of automobile weights resulting from the candidate engine is included on Table 8-15. The resulting vehicle weight for each gas turbine system is within the range of 3700 ±50 lb, so that performance and fuel consumption results, based on a vehicle weight of 3700 lb (+300 lb for passengers and cargo) for each engine, were sufficiently accurate.

TABLE 8-10

ACCESSORIES - AUTOMOTIVE GAS TURBINE ENGINE AND POWER TRANSMISSION

	IGNITION LEAD		IGNITION COIL		REGENERATOR DRIVE MOTOR		WATER INJECTION TANK		WATER INJECTION VALVE		STARTER		AIR INLET FILTER		LUBRICATING OIL		BATTERY		OIL COOLER		OIL FILTER 25 M ABS.		ALTERNATOR		TOTAL		
	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	
SINGLE SHAFT ENGINE WITH ROTARY REGENERATOR SKP 25192	0.15	0.30	1.0	2.90	1.5	4.50	2.0	1.00	.35	0.50	8.5	17.00	1.0	2.50	-	-	45.0	7.00	1.0	2.00	.20	3.32	7.0	20.00	67.70	61.02	
SINGLE SHAFT ENGINE PLATE FIN RECUPERATOR SKP 25196	0.15	0.30	1.0	2.90	-	-	2.0	1.00	.35	.50	8.5	17.00	1.0	2.50	-	-	45.0	7.00	1.0	2.00	.20	3.32	7.0	20.00	66.20	56.52	
SINGLE SHAFT ENGINE CERAMIC COUNTERFLOW RECUPERATOR SKP 25202	0.15	0.30	1.0	2.90	-	-	2.0	1.00	.35	.50	8.5	17.00	1.0	2.50	-	-	45.0	7.00	1.0	2.00	.20	3.32	7.0	20.00	66.20	56.52	
FREE TURBINE ENGINE ROTARY REGENERATOR SKP 25189	0.15	0.30	1.0	2.90	1.5	4.50	-	-	-	-	8.5	17.00	1.5	3.75	-	-	45.0	7.00	1.0	2.00	.20	3.32	7.0	20.00	65.85	60.77	
VARIABLE SPEED TRACTION TRANSMISSION SKP 25190															7.4	3.20											SAMOTRAC 50 1.25/QT - CANS .80/QT - BULK
VARIABLE SPEED BELT TRANSMISSION SKP 25188															7.8	5.00											MIL-L-7808 OIL 1.40/QT - CANS 1.25/QT - BULK

TABLE 8-11

FREE-TURBINE ENGINE ROTARY REGENERATOR SYSTEM

	ENGINE		ACCESSORIES		STAND. TRANSMISSION 3 SPEED AUTOMATIC		HYDRO. MECHANICAL CONTROLS		ELECTRONIC CONTROLS		TOTAL		
	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	
FREE TURBINE ENGINE ROTARY REGENERATOR SKP 25189 OPTION A	339.67	819.71	65.85	60.77	130.00	267.00	31.1	160.85			566.62	1308.33	
OPTION B	339.67	819.71	65.85	60.77	130.00	267.00			24.9	199.03	560.42	1346.51	

TABLE 8-12

SINGLE SHAFT ENGINE
ROTARY REGENERATOR SYSTEM

	ENGINE		ACCESSORIES		VARIABLE SPEED TRACTION TRANSMISSION SKP 25190*		VARIABLE SPEED BELT TRANSMISSION SKP 25188 Δ		HYDRO-MECHANICAL CONTROLS		ELECTRONIC CONTROLS		TOTAL		
	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	
SINGLE SHAFT ENGINE ROTARY REGENERATOR SKP 25192 OPTION A	256.35	428.71	67.70	61.02	136.31	233.40			21.50	137.22			481.86	860.35	
OPTION B	256.35	428.71	67.70	61.02	136.31	233.40					17.40	158.80	477.76	881.93	
OPTION C	256.35	428.71	67.70	61.02			132.27	196.67	21.50	137.22			477.82	823.62	
OPTION D	256.35	428.71	67.70	61.02			132.27	196.67			17.40	158.80	973.72	845.20	

*Includes $3.20 (4 qts) Sanotrac 50 lubricating oil

ΔIncludes $5.00 (4 qts) MIL-L-7808 lubricating oil

TABLE 8-13

SINGLE-SHAFT ENGINE CERAMIC COUNTERFLOW RECUPERATOR SYSTEM

	ENGINE		ACCESSORIES		VARIABLE SPEED TRACTION TRANSMISSION SKP 25190*		VARIABLE SPEED BELT TRANSMISSION SKP 25188 Δ		HYDRO-MECHANICAL CONTROLS		ELECTRONIC CONTROLS		TOTAL		
	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	
SINGLE SHAFT ENGINE CERAMIC COUNTERFLOW RECUPERATOR SKP 25202															
OPTION A	310.0	479.61	66.20	56.52	136.31	233.40			21.50	137.22			534.01	906.75	
OPTION B	310.0	479.61	66.[illegible]	56.52	136.31	233.40					17.40	158.80	529.91	928.33	
OPTION C	310.0	479.61	66.20	56.52			132.27	194.67	21.50	137.22			529.97	868.02	
OPTION D	310.0	479.61	66.20	56.52			132.27	194.67			17.40	158.80	525.87	889.60	

*Includes $3.20 (4 qts) Sanotrac 50 lubricating oil

ΔIncludes $5.00 (4 qts) MIL-L-7808 lubricating oil

TABLE 8-14

SINGLE-SHAFT ENGINE PLATE-FIN RECUPERATOR

	ENGINE		ACCESSORIES		VARIABLE SPEED TRACTION TRANSM. SKP 25190*		VARIABLE SPEED BELT TRANSMISSION SKP 25188		HYDRO-MECHANICAL CONTROLS		ELECTRONIC CONTROLS		TOTAL		
	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	LB	$	
SINGLE SHAFT ENGINE PLATE FIN RECUPERATOR SKP 25196															
OPTION A	337.29	525.51	66.20	56.52	136.31	233.40			21.50	137.22			561.30	952.65	
OPTION B	337.29	525.51	66.20	56.52	136.31	233.40					17.40	158.80	557.20	974.23	
OPTION C	337.29	525.51	66.20	56.52			132.27	196.67	21.50	137.22			557.26	915.92	
OPTION D	337.29	525.51	66.20	56.52			132.27	196.67			17.40	158.80	553.16	937.50	

*Includes $3.20 (4 qts) Sanotrac 50 lubricating oil

ΔIncludes $5.00 (4 qts) MIL-L-7808 lubricating oil

TABLE 8-15

WEIGHT SUMMARY, FULL-SIZE, SIX-PASSENGER AUTOMOBILE, POUNDS

Engine Transmission Configuration	Engine (including control system)	Transmission	Exhaust	Remainder*	Totals
V-8 Spark Ignition, Automatic	565	160	55	3135**	3915
Free-Turbine, Automatic	437	130	100	3085	3752
Single Shaft, Regenerated, Water Injection, Traction	344	136	100	↓	3665
Belt	346	132	100	↓	3663
Single Shaft, Recuperated Belt, Water Injection Metal	425	132	100	↓	3742
Ceramic	398	132	100	↓	3715

*Includes body, frame, suspension, driveline, brakes, fuel system, steering, climate control, gages and electrical circuitry. Weights supplied by OAP.

**Includes 50-lb battery; battery included in engine weight for gas turbines.

SECTION 9

ECONOMIC ANALYSIS

9.1 CONSUMER INITIAL COST

A comparison of consumer initial costs for a full-size, six-passenger automobile with a standard V-8 engine and the five most promising gas turbine engine configurations is presented on Table 9-1. Consumer costs were obtained from total manufacturing costs, using the methods outlined in Appendix 5 of this report. To illustrate, consumer cost of the single-shaft regenerated engine with belt transmission is obtained as follows:

$$\text{Consumer cost} = (\$1974)(1.2)^3(1.04) + \$176 = \$3723$$

9.2 REPAIR AND MAINTENANCE EXPENSES

Estimated repair and maintenance requirements for present Otto engine-type vehicles are shown on Tables 9-2 and 9-3. These estimates are based on factory-recommended maintenance schedules and, commonly known, approximate miles-to-failure for engine and transmission components. Note that the 7-year total (105,000 mi) of $1522 is very near the value for the expense reported in the Bureau of Census Statistical Abstracts (Appendix 6).

The corresponding estimates for the five gas turbine systems are shown on Tables 9-4 and 9-5. (Only four systems are listed on Table 9-5, since the repair and maintenance requirements for the two recuperated, single-shaft engines are expected to be about equal, whether the recuperator material used is metal or ceramic.)

TABLE 9-1

SUMMARY OF ESTIMATED MANUFACTURING AND CONSUMER COSTS, DOLLARS
(FULL-SIZE, SIX-PASSENGER, FOUR-DOOR SEDAN, WITHOUT AIR-CONDITIONING)

Engine-Transmission Configuration	Engine	System Controls	Accessories	Transmission	Drive Line, Body, Frame, Interior Wheels, Tires, and Misc.	Total Manufacturing Cost	Total Consumer Cost
V-8 Spark Ignition Automatic (1970 List)	356*	--	--	267	1157	1780	3185
Traction	356	--	--	233	1157	1746	3130
Free-Turbine, Regenerated Automatic	820	161	61	267	1150	2459	4595
Single-Shaft, Regenerated Water Injection Traction	429	137	↓	233	↓	2010	3788
Belt	429	↓	↓	197	↓	1974	3723
Recuperated Belt							
Metal	526	↓	57	↓	↓	2067	3890
Ceramic	480	↓	57	↓	↓	2021	3808

*Includes controls and engine-mounted accessories

TABLE 9-2

ESTIMATED MAINTENANCE REQUIRED PER YEAR, V-8 SPARK-IGNITION ENGINE (15,000 MI/YR, 7-YR LIFE)

Maintenance Item	Interval	Average Cost for Each Service, Dollars	Cost per Year, Dollars
Chassis Lubrication	4 mo or 6000 mi	3.50	10.50
Engine Oil and Filter	4 mo or 6000 mi	7.50	22.50
Air Filter	1/yr	3.50	3.50
Rotate Tires	7,500 mi	3.00	6.00
Pack Wheel Bearings	15,000 mi	4.00	4.00
Brake Fluid, PCV, Fuel Filter, Differential Lube	1/yr	3.00	3.00
Wash and Wax	10/yr	2.00	20.00
Total Cost/Year			69.50

TABLE 9-3

REPAIR AND MAINTENANCE ESTIMATE FOR 105,000 MILES, 7-YEAR LIFE, V-8 SPARK-IGNITION ENGINE (BASED ON 1970 ENGINE AND AUTOMOBILE)

Maintenance or Repair Item	Average Interval, mi	Average Cost for Each Repair or Service, Dollars	Years in Which Repair or Maintenance Required	Cost/Year, Dollars							Seven Year Total
				First	Second	Third	Fourth	Fifth	Sixth	Seventh	
Front end alignment	20,000	12.00	2, 3, 4, 6, 7	--	12.00	12.00	12.00	--	12.00	12.00	
Engine tune-up	20,000	40.00	2, 3, 4, 6, 7	--	40.00	40.00	40.00	--	40.00	40.00	
Transmission service	24,000	26.00	2, 4, 5, 7	--	26.00	--	26.00	26.00	--	26.00	
Flush and fill radiator	2 yr	15.00	2, 4, 6	--	15.00	--	15.00	--	15.00	--	
Reline brakes	30,000	50.00	2, 4, 6	--	50.00	--	50.00	--	50.00	--	
Replace radiator hose	3 yr	5.00	3, 6	--	--	5.00	--	--	5.00	--	
Replace water pump	40,000	24.00	3, 6	--	--	24.00	--	--	24.00	--	
Replace battery	3 yr	30.000	3, 6	--	--	30.00	--	--	30.00	--	
Grind valves and replace rings	60,000 - 80,000	250.00	5	--	--	--	--	250.00	--	--	
Starter brushes and commutator	50,000	15.00	4	--	--	--	15.00	--	--	--	
Replace alternator	60,000	30.00	4	--	--	--	30.00	--	--	--	
Maintenance yearly (Table 9-2)	15,000	69.50	1, 7	69.50	69.50	69.50	69.50	69.50	69.50	69.50	
Miscellaneous (alternator belt, wiper)	15,000	--	1, 7	3.50	6.00	10.00	12.00	12.00	10.00	10.00	
TOTAL				73.00	218.50	190.50	269.50	357.50	255.50	157.50	1522.00

TABLE 9-4

ESTIMATED MAINTENANCE REQUIRED PER YEAR, GAS TURBINE ENGINE (15,000 mi/yr, 7-yr life)

Maintenance Item	Average Interval	Average Cost for Each Service	Cost per Year
Chassis Lubrication	4 mo or 6000 mi	$3.50	$10.50
Air Filter	1 per yr	5.50	5.50
Rotate Tires	7,500 mi	3.00	6.00
Pack Wheel Bearings	15,000 mi	4.00	4.00
Brake Fluid Differential Lube, Fuel Filter	1 per yr	3.00	3.00
Oil Filter	15,000 mi	3.00	3.00
Wash and Wax	10 per yr	2.00	20.00
TOTAL			$52.00

TABLE 9-5

REPAIR AND MAINTENANCE ESTIMATE FOR GAS TURBINE ENGINES
(105,000 MILES, 7-YEAR LIFE)

Maintenance or Repair Item	Average Interval, mi	Average Cost for Each Repair or Service	Years In Which Repair or Maintenance Required	First Year	Second Year	Third Year	Fourth Year	Fift Year	Sixth Year	Seventh Year	Seven Year Total
Front End Alignment	20,000	$12.00	2, 3, 4, 6, 7	--	$ 12.00	$ 12.00	$ 12.00	--	$ 12.00	$ 12.00	
Reline Brakes	30,000	50.00	2, 4, 6	--	50.00	--	50.00	--	50.00	--	
Battery	3 yr	30.00	3, 6	--	--	30.00	--	--	30.00	--	
Starter Brushes and Commutator	50,000	15.00	4	--	--	--	15.00	--	--	--	
Alternator	60,000	30.00	4	--	--	--	30.00	--	--	--	
Ignition Coil, Lead and Ignitor	55,000	15.00	4	--	--	--	15.00	--	--	--	
(1)(2)(3) Transmission Fluid	55,000	4.50	4	--	--	--	(1)(2)(3) 4.50	--	--	--	
(2)(3) Transmission Belt	25,000	12.50	2, 4, 5, 7	--	(2)(3) 12.50	--	(2)(3) 12.50	(2)(3) 12.50	--	(2)(3) 12.50	
(1)(2)(4) Regenerator Seals	55,000	27.50	4	--	--		(1)(2)(4) 27.50	--	--	--	
(1)(2)(3) Water Hose	3 yr	5.00	3, 6	--	--	(1)(2)(3) 5.00	--	--	(1)(2)(3) 5.00	--	
Transmission (4) Service	24,000	26.00	2, 4, 5, 7	--	(4) 26.00	--	(4) 26.00	(4) 26.00	--	(4) 26.00	
Maintenance Yearly Table 9-4	15,000	52.00	1, 7	52.00	52.00	52.00	52.00	52.00	52.00	52.00	
Miscellaneous (Alternator Belt, Wiper Blades, etc.)	15,000	--	1, 7	3.50	6.00	10.00	12.00	12.00	10.00	10.00	
TOTALS, DOLLARS											
(1) Single-Shaft, Regenerated, Water Injection, Traction Transmission				57.50	120.00	109.00	218.00	64.00	159.00	74.00	801.50
(2) Single-Shaft, Regenerated, Water Injection, Belt Transmission				↓	132.50	↓	230.50	76.50	↓	86.50	851.50
(3) Single-Shaft, Recuperated, Water Injection, Belt Transmission				↓	132.50	↓	203.00	76.50	↓	86.50	824.00
(4) Free-Turbine, Regenerated, Three-Speed Automatic Transmission				↓	146.00	104.00	239.50	90.50	154.00	100.00	891.00

9.3 COST OF OWNERSHIP

Results of estimated cost-of-ownership values for an automobile with the spark-ignition engine and with the five gas turbine systems are listed on Table 9-6. Elements of the total cost-of-ownership are approximately the same as those in Appendix 6. Fuel cost for the spark-ignition engine was based on an average consumption of 14.8 mpg, as computed in this study, rather than the 13.8 mpg shown in Appendix 6.

Fuel cost for a gas turbine engine differs from that for a spark-ignition engine in that a gas turbine may use JP and diesel fuels rather than the low-lead, high-octane that will be required by 1975 or before. Although fuel prices vary widely with location and time, present distributor cost of JP fuels is at least $0.02/gal less than regular leaded gasoline of 90-96 octane rating. Most of this cost differential is in tetraethyl lead. Thus, if present fuel costs were used in the cost-of-ownership analysis, a differential of $0.02/gal should be assumed.

If large quantities of JP fuel were demanded, distributor costs would have to be increased to cover amortization of the capital equipment required. However, at that time, the cost of nonleaded 90-96 octane gasoline would be increased by about $0.02/gal, as estimated by oil industry representatives. Thus, the cost differential between JP fuels and gasoline would still remain at about $0.02/gal.

In this analysis, fuel costs were assumed to be $0.34 and $0.36/gal for JP and gasoline fuels, respectively. Also, the heat release per gallon of JP fuel is necessarily greater than that for high octane gasoline because of the higher densities of the JP fuels. For example, JP-5 has a heat release of about 12.5×10^4 Btu/gal compared with 11×10^4 Btu/gal for gasoline, or a difference of about 11 percent. Thus, the gasoline-burning engine would have to have an 11-percent

TABLE 9-6

COST-OF-OWNERSHIP COMPARISON, DOLLARS
(FULL-SIZE, SIX PASSENGER AUTOMOBILE; 105,000-MILE, 7-YEAR LIFE)

Engine-Transmission Configuration	Depreciation (Zero Scrap Value)	Repairs and Maintenance	Replacement Tires and Tubes	Fuel and Water Cost**	Oil	Insurance	Garaging, Parking, Tolls*	Taxes (Excluding Gas Tax)	Total Cost of Ownership	Cost of Ownership, Percent of First System
V-8 Spark Ignition, Automatic	3185	1522	385	2560	158	1722	1805	557	11,894	100
Traction	3130	1422	↓	1980	158	↓	↓	↓	11,159	94
Free-Turbine, Regenerated, Automatic	4595	8919		1800	--				11,755	99
Water Injection	4595	891		1700	--				11,655	98
Single-Shaft, Regenerated, Water Injection Traction	3788	802		1773	--				10,832	91
Belt	3723	851		1670	—				10,713	90
Recuperated Belt Metal	3890	824		1842					11,025	93
Ceramic	3808	↓		1842	—				10,943	92
Traction	3808			2140					11,241	95

*See Appendix 5

**Based on costs of $0.35/gal. for JP-4 fuel (including local and Federal taxes and $0.01/gal. of fuel for water) and $0.36/gal. for leaded 90-96 octane gasoline.

better average fuel economy than the JP-burning engine to have the same total fuel cost, assuming the fuel costs per gallon were equal. The full advantage of this difference between the spark-ignition and gas turbine engines was not used in this study. Here, it was assumed that the fuel had a volumetric heat release of 11.58×10^4 Btu/gal for JP fuels and 11.0×10^4 Btu/gal for gasoline.

In Appendix 2, an estimate was made that a water-injected engine would consume about 0.5 gal of water for each gallon of fuel and that the cost of the de-ionized water would be about $0.01/gal. Thus, the cost of fuel should be increased by $0.005/gal to account for the water cost. For water-injected engines, a somewhat conservative fuel cost of $0.35/gal was used.

In an EPA report to Congress[11], 1975 was estimated as the year that emissions controls could increase the price of a new car by $262.50 and add an additional operating and maintenance cost of $20.70 per year, or a total additional cost-of-ownership of $407.40. This would increase the total cost-of-ownership for the automobile with spark-ignition engine to $12,490. This is approximately $900 greater than the most expensive gas turbine and about $1800 more than the least expensive.

SECTION 10

RECOMMENDED CONFIGURATION

The five engine transmission systems were rated on a scale of 1 to 5, with respect to each of the optimizing parameters (Table 10-1). The higher rating values indicate a greater margin, relative to other systems, in meeting the design objective for the given parameter. Thus, the system having the greatest total is the optimum for the automobile. Each of the five engines have been previously analyzed with respect to the parameters listed on Table 10-1. The selection of the rating values are logical but not entirely obvious and, therefore, will be briefly discussed.

The ratings for emissions are based on Table 7-9, on which is shown Systems 4 and 5 as slightly better than 2 and 3, and considerably better than System 1.

Cost-of-ownership ratings are obtained from Table 9-6. Systems 2, 3, 4, and 5 have values that range from 89 to 91 percent of the value estimated for the vehicle with a spark-ignition engine and are considered equal, within the accuracy of the prediction procedures. The cost-of-ownership for System 1 is high enough to warrant a slightly lower rating. All systems have an estimated cost-of-ownership less than that for the spark-ignition engine and were given high ratings accordingly.

The engine response values are based on the times required to accelerate the engine from idle to a speed corresponding to maximum output torque, as listed on Tables 7-3, 7-4, and 7-5. Only Systems 2 and 3 clearly meet the requirement; Systems 4 and 5 are marginal, and System 1 exceeds the required time, for reasons discussed in Section 7. This parameter is less important than the others; hence, the best system is limited to a maximum rating of 4.

TABLE 10-1

COMPARISON OF CANDIDATE SYSTEMS WITH RESPECT TO DESIGN OBJECTIVES

Engine Transmission System	Emissions	Cost of Ownership	Engine Acceleration Performance	Maintainability and Reliability	Initial Cost	Engine Mechanical Design Margin**	Total
1. Free-Turbine, Regenerated, Three-Speed Automatic	2*	3	2	2	1	4	14
2. Single-Shaft, Regenerated, Water Injection, Traction	4	5	4	5	2	5	25
3. Belt	4	5	4	3	2	5	23
4. Single-Shaft, Recuperated, Water Injection, Belt Metal	5	5	3	4	2	3	22
5. Ceramic	5	5	3	4	2	3	22

*Systems are rated on a 1-5 scale; larger numbers mean greater margin in meeting design objective.

**Based on turbine life and stress levels. Other mechanical features are approximately equal for all engines.

Maintainability ratings were obtained from Table 9-5. The order or these ratings is based on an assumption that the regenerator seals can be designed to provide a life of 55,000 mi and have to be replaced only once. This assumption was not justified by analysis or data reported in the literature.

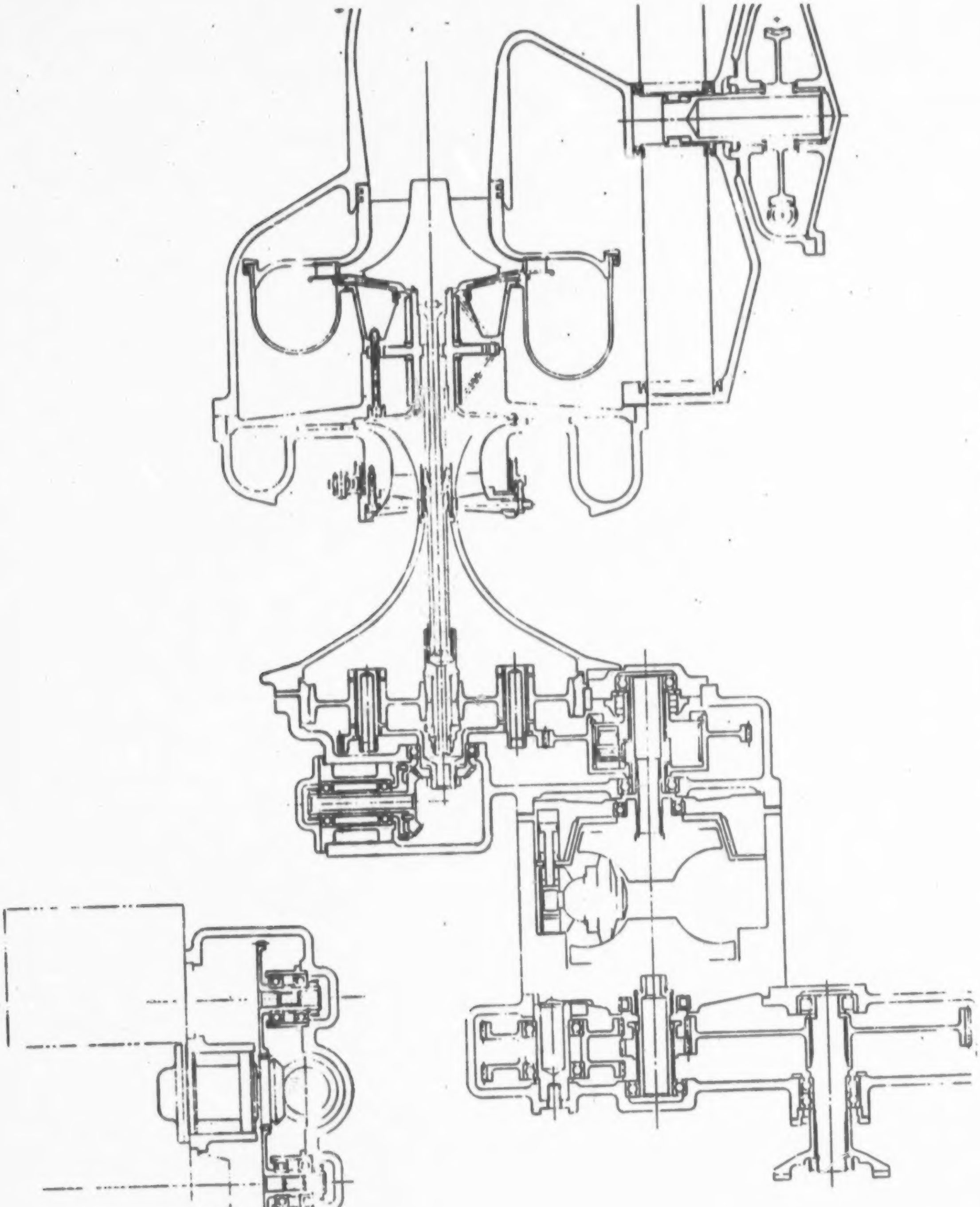
Initial cost ratings were based on total consumer cost values shown on Table 9-1. There is little difference between initial costs for Systems 2 through 5, relative to the difference between these and System 1. Also, all systems provide a consumer initial cost greater than that for the spark-ignition engine and, therefore, were given ratings no higher than 2.

The totals then clearly identify the single-shaft engine as the optimum gas turbine for the automobile. However, one of the four single-shaft engine systems has not a distinct advantage over the other three. Although the belt transmission provides a slightly better system rating, the traction drive might be favored since it has already undergone significant development for constant-speed drive applications with poorer tractive effort fluids than the SANTOTRAC fluid. However, after further development, the belt drive may prove to be the best transmission because of the expected high efficiency and potential low cost. If emissions were the only criteria, the recuperated engine would be chosen, based on limited data and analytical techniques, but if heat exchanger bypass air were used in the combustor primary zone, the regnerated engine might be more desirable because of the lower pressure ratio and, hence, lower compressor discharge temperature. Therefore, both the regenerated and recuperated single-shaft engines are recommended to be prototype-tested to the point where one has the distinct advantage. Similarly, both the traction and the belt transmission should be further tested before making a final decision.

An assembly drawing of one of the recommended engine-transmission systems is included as Figure 10-1. The installation of this system in a standard-sized U.S. automobile is shown in Figure 10-2. Either of the other three systems would package in a similar manner.

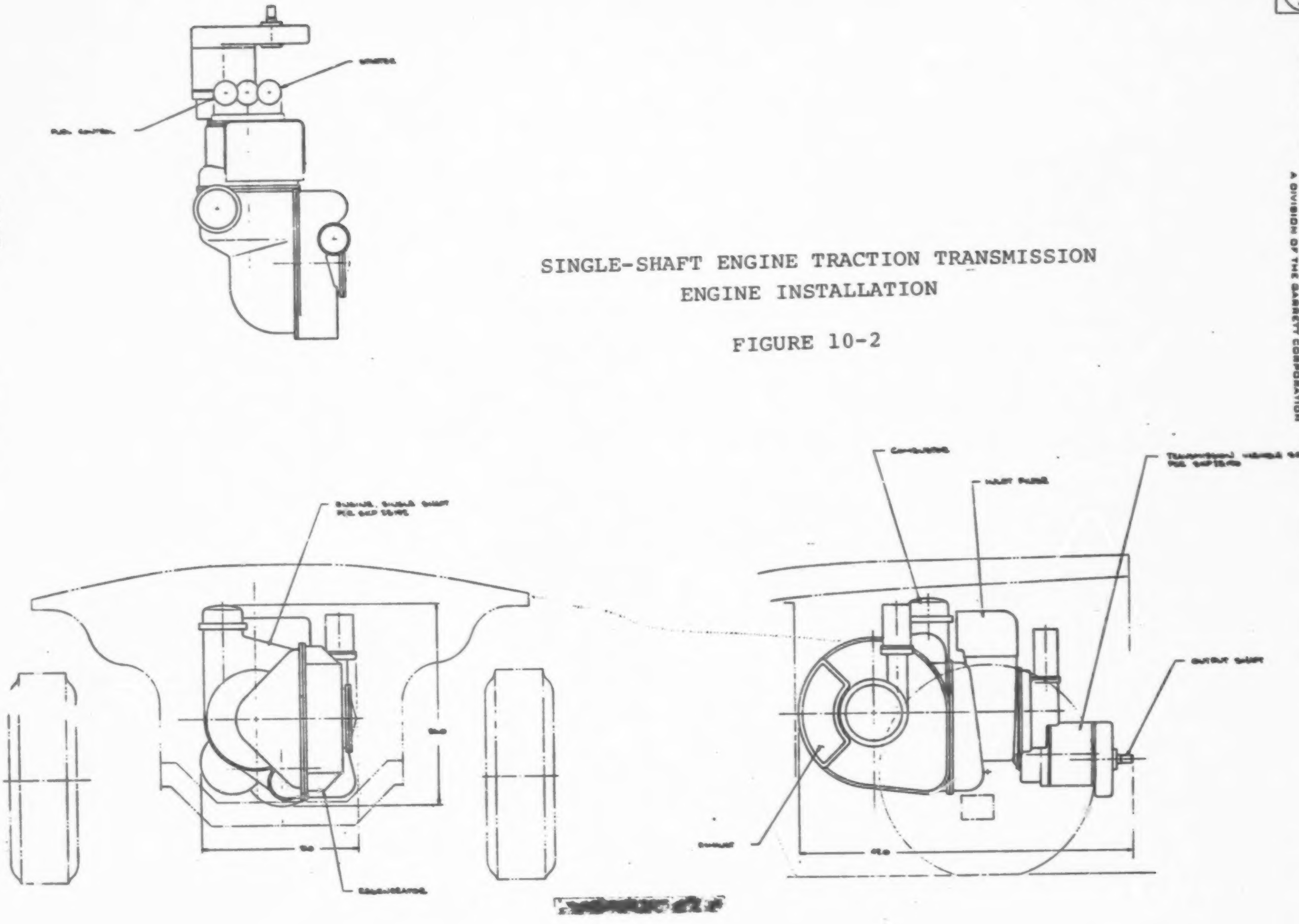
A schematic of the recommended configuration is shown in Figure 10-3. Where two numbers are shown (such as, for output power, component efficiencies, and pressure ratios), the first number applies to the recuperated engine and the second number to the regenerated engine. These engines have variable inlet guide vanes that vary continuously--- a single-stage centrifugal compressor, and a single-stage radial flow turbine. The compressor and turbine efficiencies are total-to-total values. Both engines have exhaust heat recovery with effectiveness values of 0.85 for the recuperated engine and 0.90 for the regenerated engine. The numbers shown in parentheses are pressure loss values, and the numbers in brackets are leakage losses, both given in percent. The 5.4-percent leakage loss in the heat exchanger applies only to the regenerated engine, as the regenerator seal leakage.

The sea-level, standard-day, output power ratings are 165 and 155 hp for the recuperated and regenerated engines, respectively. These are the power ratings required to meet all vehicle acceleration performances. The rated power can be reduced by using power augmentation for hot-day, maximum power accelerations. Water injection is the best means to achieve this power boost. One engine has demonstrated a 40-percent power increase on a 100°F day, by injecting water at the compressor inlet at a rate of 6 percent water-to-air on a mass ratio basis. In the analysis, a 30-percent power boost was used for the same 6-percent water injection rate. Using power augmentation reduces the engine size and cost, increases fuel economy by about 2 mpg on the FDC and on the composite route, decreases emissions, and reduces vehicle cost-of-ownership. However, some of these claims may be made for water injection of spark-ignition and compression ignition engines, but they are not seriously considered for these engines, probably



SINGLE REGENERATOR TRACTION TRANSMISSION
SINGLE-SHAFT ENGINE

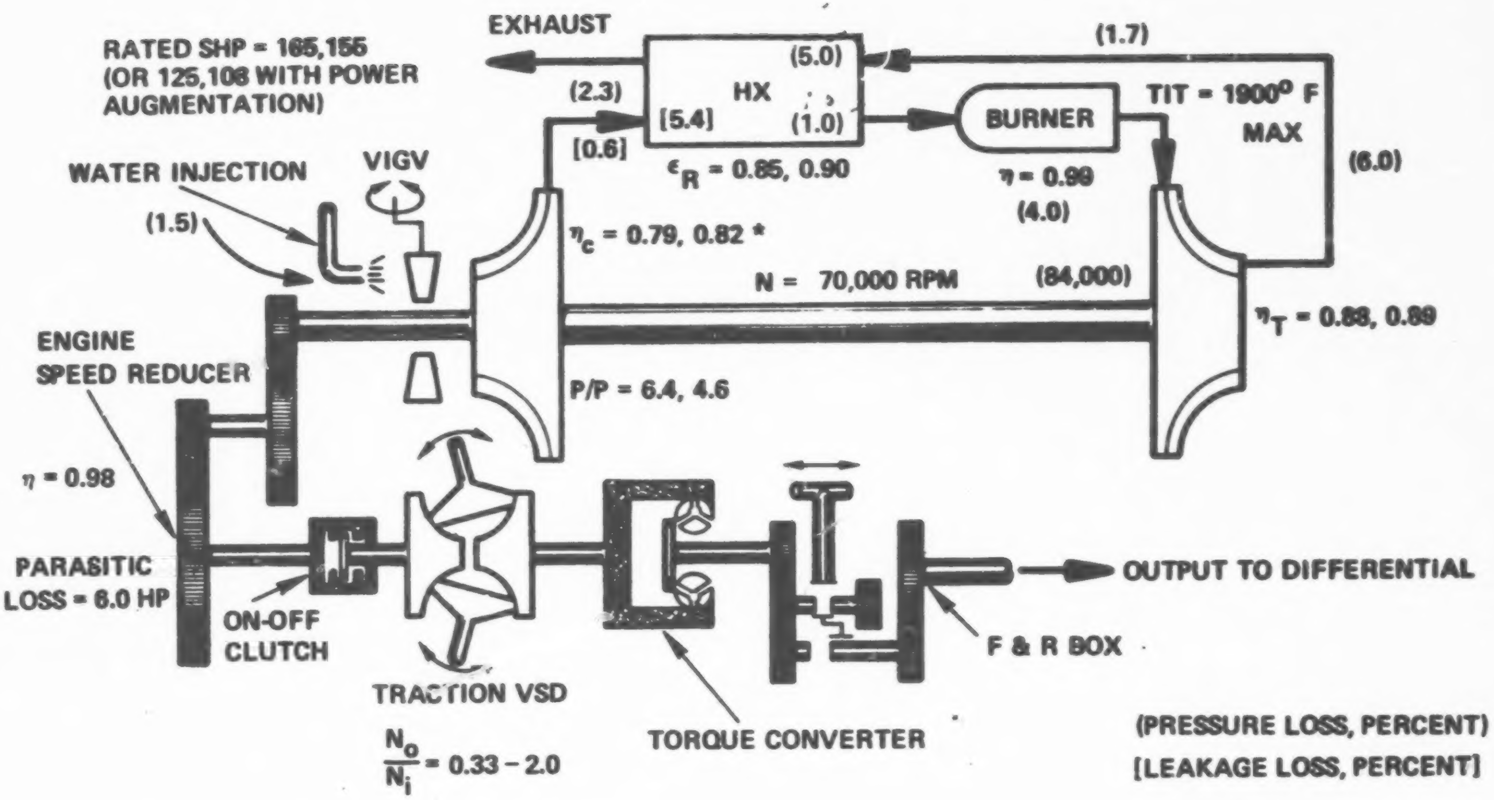
FIGURE 10-1



SINGLE-SHAFT ENGINE TRACTION TRANSMISSION
ENGINE INSTALLATION

FIGURE 10-2

[redacted]


RATED SHP = 165,155
(OR 125,108 WITH POWER
AUGMENTATION)
EXHAUST
WATER INJECTION
(1.5)
VIGV
(2.3)
[0.6]
(5.0)
HX
[5.4]
(1.0)
ϵ_R = 0.85, 0.90
(1.7)
BURNER
η = 0.99
(4.0)
TIT = 1900° F
MAX
(6.0)
η_c = 0.79, 0.82 *
N = 70,000 RPM
(84,000)
η_T = 0.88, 0.89
ENGINE
SPEED REDUCER
P/P = 6.4, 4.6
η = 0.98
PARASITIC
LOSS = 6.0 HP
ON-OFF
CLUTCH
TRACTION VSD
$\frac{N_o}{N_i}$ = 0.33 - 2.0
TORQUE CONVERTER
F & R BOX
OUTPUT TO DIFFERENTIAL
(PRESSURE LOSS, PERCENT)
[LEAKAGE LOSS, PERCENT]

*WHERE TWO NUMBERS ARE GIVEN, THE FIRST NUMBER APPLIES TO THE RECUPERATED CYCLE AND THE SECOND NUMBER TO THE REGENERATED CYCLE

SINGLE-SHAFT ENGINE - SYSTEM SCHEMATIC

FIGURE 10-3

because of logistics, safety, and practicality. Therefore, use of the non-power-boosted engines is suggested in making comparisons with other gas turbines and other power systems for the automobile.

Output power is extracted at the compressor end of these engines. A two-stage spur gear set reduces the engine speed to an acceptable level for the variable speed drive. A planetary gear set could also be used for the speed reducer.

The on-off clutch is disengaged only during engine start-up to allow lubrication pressure to build up for the hydrostatic thrust bearings in the traction drive, prior to rotating the unit. For the traction element, the Tracor traction drive was used with an overall speed ratio of 6:1, consisting of a 2:1 speed-up and a 3:1 speed-down. The output shaft of the Tracor unit drives a conventional torque converter and a forward and reverse gearbox.

SECTION 11

PROGRAM PLANS - RECOMMENDED DEVELOPMENT AND DEMONSTRATION PROGRAM

To achieve successful demonstration of the recommended single-shaft engine-transmission system described in this study, it is necessary to prove experimentally that the system provides vehicular emissions that are considerably lower than the 1976 Federal Standards, that the vehicle has acceleration performance, fuel economy, and handling characteristics as good as, or better than, a 1970 automobile, and that the vehicle can be maintained by the consumer without increased attention to adjustments, tuning, or diagnostic system checks. These objectives must be met without significantly increasing the complexity of the system to meet the emissions requirements, otherwise the initial cost of the system may increase beyond practical limits.

The components requiring the most development effort to achieve successful demonstration of the recommended system are the combustor, the foil-type air bearings for the hot section of the engine, and the traction variable-speed drive.

11.1 PROGRAM LOGIC CHART

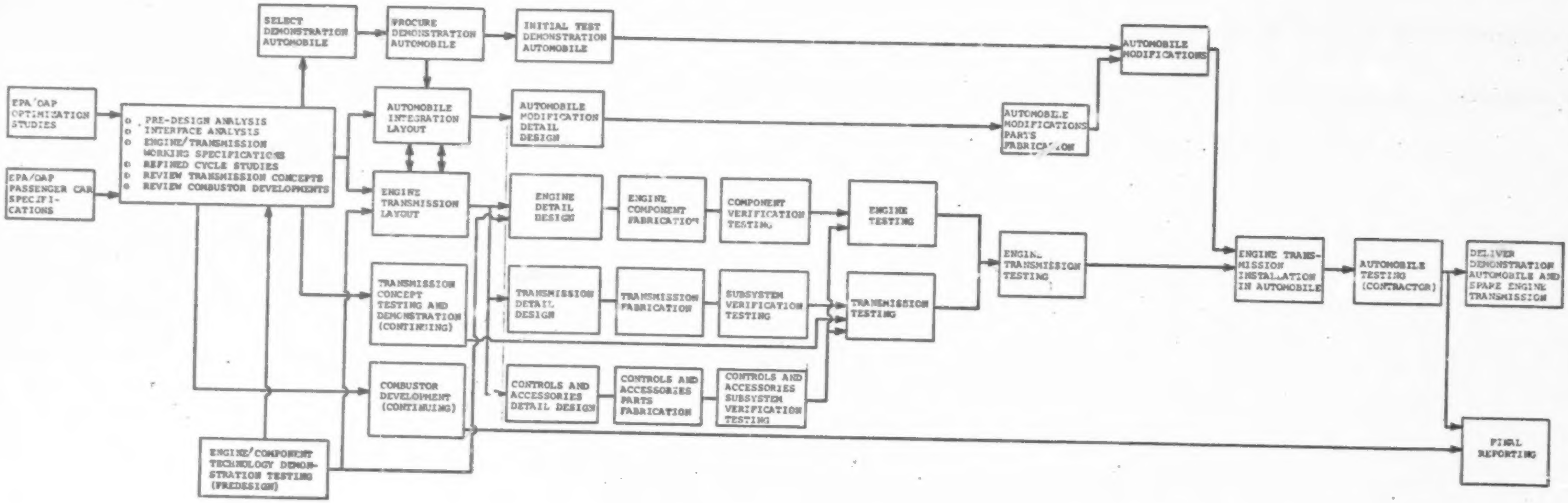
A diagram of the minimum cost development program logic for the recommended system is shown in Figure 11-1. This program is defined as that required to demonstrate one version of the recommended system, i.e., the single-shaft recuperated (metallic) engine with a traction transmission. The minimum program excludes such items as to determine and demonstrate the optimum transmission (traction, hydromechanical, or belt) and to determine the optimum heat-recovery device (regenerator or recuperator). The minimum program also assumes that the system is developed only to an extent where the system can be demonstrated in an automobile and is not necessarily ready for extensive endurance tests or fleet demonstrations.

The results of a study and demonstration program to determine the optimum heat-recovery device, fixed-boundary recuperator, or rotary regenerator should be completed prior to beginning the design phase of the program. However, to meet the end date of the demonstration program, it may be necessary to perform the heat-exchanger trade-off program in parallel with the system demonstration program. This could be accomplished without major schedule impact on the overall program by designing the power section to accept either type of heat exchanger. Also, a transmission trade-off study and demonstration program should be conducted to select the optimum transmission prior to finalizing the optimum system design. Again, this task could be performed in parallel with the demonstration program without greatly extending the time required for the program.

The heat exchanger and transmission programs would increase the total cost of the program by about 3.5 million dollars (depending on the scope) but would considerably reduce the risk of not meeting all of the program objectives in the specified time.

Results of the EPA-sponsored study and design programs for the engine, combustor, and transmission would be used to define details of the engine-transmission system. This would be designed to meet all requirements in the EPA performance specification.[12]

Refined cycle studies would be performed, to account for modifications in the system, such as increased combustor pressure drop or limitations on combustor inlet temperature to achieve the emissions requirements. Engine and transmission specifications would be written early in the program to aid the detail design of these subsystems. Mission analysis studies would be performed to make a final selection


EPA/OAP OPTIMIZATION STUDIES
EPA/OAP PASSENGER CAR SPECIFICATIONS
o PRE-DESIGN ANALYSIS
o INTERFACE ANALYSIS
o ENGINE/TRANSMISSION WORKING SPECIFICATIONS
o REFINED CYCLE STUDIES
o REVIEW TRANSMISSION CONCEPTS
o REVIEW COMBUSTOR DEVELOPMENTS
ENGINE/COMPONENT TECHNOLOGY DEMONSTRATION TESTING (PREDESIGN)
SELECT DEMONSTRATION AUTOMOBILE
PROCURE DEMONSTRATION AUTOMOBILE
INITIAL TEST DEMONSTRATION AUTOMOBILE
AUTOMOBILE INTEGRATION LAYOUT
AUTOMOBILE MODIFICATION DETAIL DESIGN
ENGINE TRANSMISSION LAYOUT
TRANSMISSION CONCEPT TESTING AND DEMONSTRATION (CONTINUING)
COMBUSTOR DEVELOPMENT (CONTINUING)
ENGINE DETAIL DESIGN
ENGINE COMPONENT FABRICATION
COMPONENT VERIFICATION TESTING
TRANSMISSION DETAIL DESIGN
TRANSMISSION FABRICATION
SUBSYSTEM VERIFICATION TESTING
CONTROLS AND ACCESSORIES DETAIL DESIGN
CONTROLS AND ACCESSORIES PARTS FABRICATION
CONTROLS AND ACCESSORIES SUBSYSTEM VERIFICATION TESTING
ENGINE TESTING
TRANSMISSION TESTING
ENGINE TRANSMISSION TESTING
AUTOMOBILE MODIFICATIONS PARTS FABRICATION
AUTOMOBILE MODIFICATIONS
ENGINE TRANSMISSION INSTALLATION IN AUTOMOBILE
AUTOMOBILE TESTING (CONTRACTOR)
DELIVER DEMONSTRATION AUTOMOBILE AND SPARE ENGINE TRANSMISSION
FINAL REPORTING

ADVANCED GAS TURBINE AUTOMOBILE MINIMUM COST DEMONSTRATION PROGRAM LOGIC CHART

FIGURE 11-1

of engine power rating, to compute fuel economy, and to estimate FDC emissions. The predesign analysis effort would include critical speed and stress analyses, flow-path design, thermal analysis of potential configurations, and the selection of a final engine and transmission configuration. Also, control system dynamics analysis would be performed to determine parameter values for system stability and desired response.

The demonstration automobile would be selected early in the program to define the envelope dimensions of the engine compartment, to specify the engine-mounting points, and to design inlet and exhaust ducting subsystems. Initial tests of the demonstration automobile would include acceleration performance, fuel economy, and emissions tests with the conventional standard-size engine.

Tests and demonstrations of an advanced recuperator and a low-emission combustor on an existing AiResearch engine would be used to aid definition of these components in the engine design.

Detail layout designs of the engine, transmission, and vehicle installation of the system would be initiated. Transmission and combustor component testing would commence early in the program, since test hardware already exists for these components. Data obtained in these early tests would be useful in influencing the later detail design of these same components.

Following the above tests, engine, transmission, and control system components would be designed in detail, built, and tested to verify mechanical and aerothermodynamic design objectives. Components to be rig-tested would be designed as early as possible in the program and early fabrication emphasized. These components include the compressor, turbine, foil bearings, heat exchanger, and controls. Verification tests of updated combustor and transmission designs would also be included in the component and subsystem tests.

Next would be the engine and transmission tests--separately, as subsystems for mechanical integrity, performance, response, and control stability. Emissions tests would be performed on the engine. The engine and transmission would be mated for dynamometer testing of the complete system, including simulated duty cycle, performance, emissions, controls, and endurance testing. Then, the system would be installed in the automobile for demonstration and final verification tests; after which, the vehicle would be delivered, and a final report on the results and recommendations of the program would be issued.

11.2 PROGRAM SCHEDULE

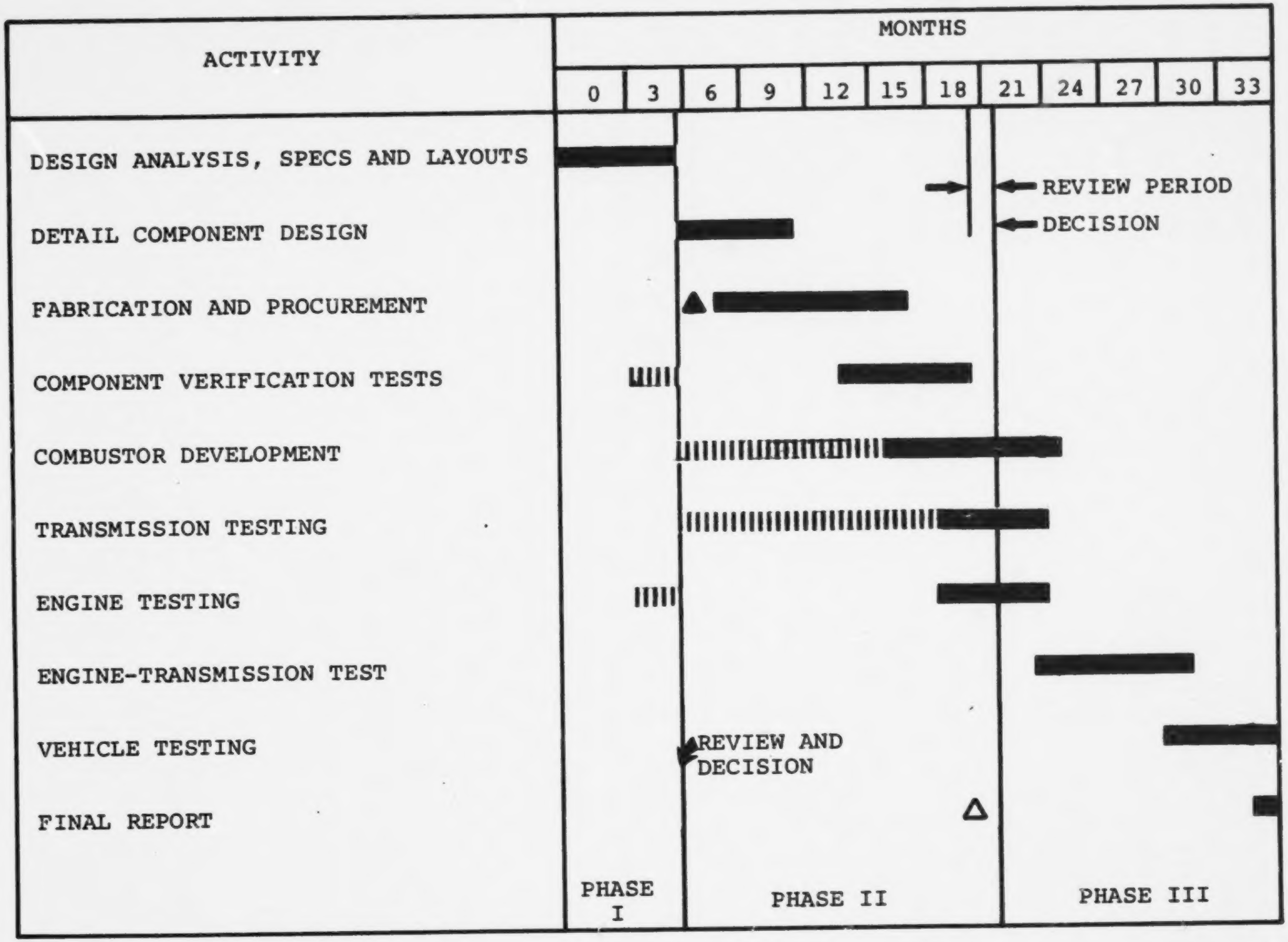
A three-phase program is proposed for achieving successful demonstration of the recommended gas turbine automobile (Figure 11-2):

Phase I consists of final-cycle studies, preliminary design layouts of the system, and demonstration of a recuperated AiResearch engine with an advanced low-emissions combustor. This phase would require 4 months for completion.

Phase II includes detail design of the system components, components verification tests, and engine and transmission subsystem tests. This phase would require 16 months for completion.

Phase III consists of engine transmission system tests, first as dynamometer and then as vehicle tests. Thirteen months would be required to complete the Phase III effort.

The complete program would require a total of 33 months for completion.



ADVANCED GAS TURBINE AUTOMOBILE DEMONSTRATION PROGRAM SCHEDULE

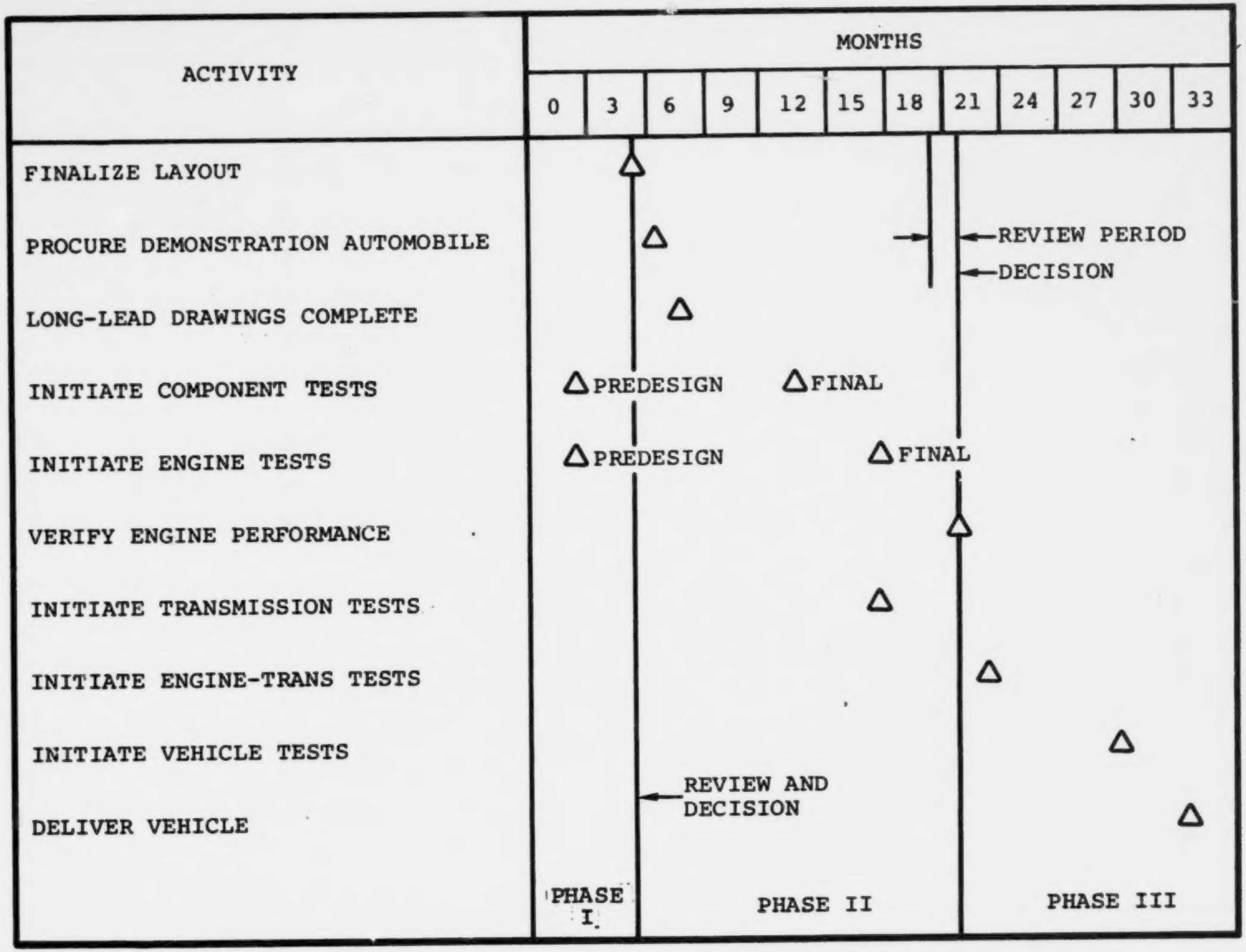
FIGURE 11-2

Significant milestones for the program are shown in Figure 11-3. Comprehensive program reviews would be held at the end of each phase. These reviews could be used as decision milestones for go-ahead on the next phase of the program.

11.3 PROGRAM COSTS

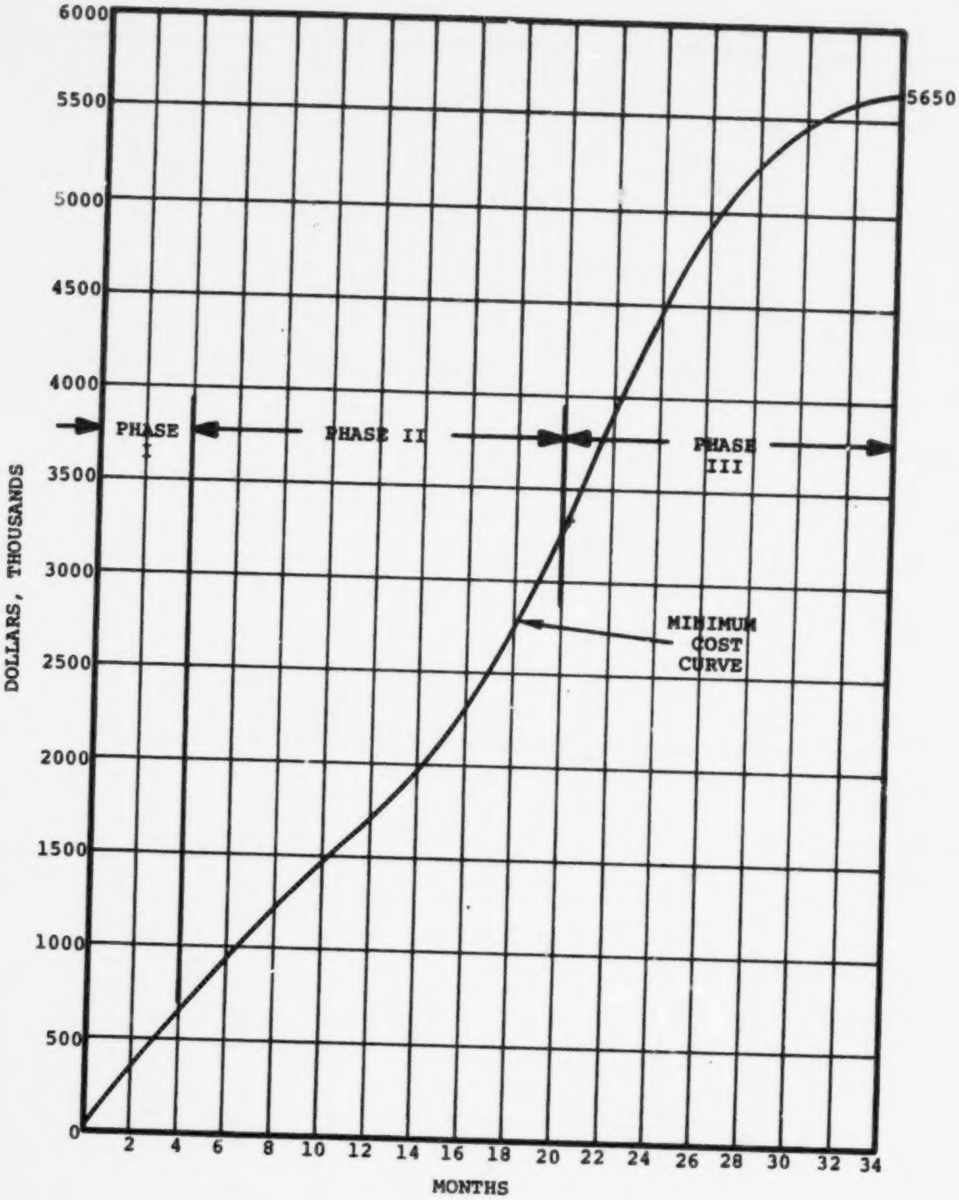
A graph of estimated minimum program total costs as a function of time is shown in Figure 11-4. The cost include expenditures for engineering and laboratory man-hours, fabrication and test hardware, and test facility time but not the general and administrative (G&A) and fee costs. A breakdown of the total costs for each of these elements is shown on Table 11-1. These also do not include general and administrative or fee costs.

A range of man-hours and dollars are provided to show the differences between the minimum program (as defined in 11.1) and the costs that might be expected for the more complete lower risk program.


ACTIVITY
MONTHS
0
3
6
9
12
15
18
21
24
27
30
33
FINALIZE LAYOUT
PROCURE DEMONSTRATION AUTOMOBILE
LONG-LEAD DRAWINGS COMPLETE
INITIATE COMPONENT TESTS
INITIATE ENGINE TESTS
VERIFY ENGINE PERFORMANCE
INITIATE TRANSMISSION TESTS
INITIATE ENGINE-TRANS TESTS
INITIATE VEHICLE TESTS
DELIVER VEHICLE
REVIEW PERIOD
DECISION
PREDESIGN
FINAL
PREDESIGN
FINAL
REVIEW AND DECISION
PHASE I
PHASE II
PHASE III

ADVANCED GAS TURBINE AUTOMOBILE DEMONSTRATION PROGRAM MILESTONES

FIGURE 11-3



PROGRAM TOTAL COSTS

FIGURE 11-4

TABLE 11-1

BREAKDOWN OF PROGRAM COSTS

Item	Hours x 10^{-3}	Dollars x 10^{-6}
Hardware		
Engines and Transmissions	-	1.62-2.90
Special Tooling and Test Equipment	-	0.51-0.85
Total Hardware		2.13-3.75
Labor		
Engineering	160-216	2.81-3.8
Laboratory	32- 74	0.52-1.2
Total Labor	190-290	3.33-5.0
Test Facility	15- 32	0.19-0.40
TOTAL Program Cost*		5.65-9.15

*Excluding G&A and Fee

SECTION 12

GAS TURBINE LONG RANGE POTENTIAL

12.1 INTRODUCTION

The purpose of this part of the program was to estimate the long range potential of the gas turbine engine with respect to initial cost, fuel economy, and cost-of-ownership for the anticipated role of a future alternate automotive powerplant. This section of the report is based on long-range, anticipated technology attainable in the near term only at far greater effort than considered reasonable for the demonstration of the optimum engine.

As discussed in a previous section, current analytical methods and experimental data for low-emissions-designed combustors do not give accurate estimates of future gas turbine engine emissions. Therefore, no attempt will be made to present these estimates in this analysis. Several companies are presently working under contracts for the OAP (reference Contract 68-04-0014 with AiResearch) on low emission combustors. Therefore, this discussion is devoted to potential advancements in technology that will aid in minimizing the remaining two parameters: fuel consumption and cost-of-ownership.

For a given vehicle, the only way to improve fuel economy is to increase the engine cycle efficiency or, equivalently, to decrease the engine specific fuel consumption (sfc). Lower sfc may be achieved by increasing component efficiencies, heat recovery effectiveness (with corresponding reduction in cycle pressure ratio), and turbine inlet temperature or, by decreasing parasitic losses, such as turbine cooling flow, flow-path pressure losses, and leakage losses.

Increased turbine inlet temperature without increased turbine cooling flow is easily one of the most promising candidates for making large improvements in cycle efficiency. However, high-temperature superalloys and coatings appear to have reached their upper limits at 1800° to 2000°F turbine inlet temperature, at a cost reasonable for automobile application. At present, the strongest contenders for this reasonable cost, high-temperature material are highly dense silicon carbide and silicon nitride.[13,14] According to R. J. Lumby (Reference 14), processing techniques are presently available for producing test parts, and sufficient production processes are known to permit reasonable estimates of material costs. However, even if optimistic results are achieved in component rig tests, considerable development work will be required before this material can be used in actual gas turbine engines.

This analysis is not necessarily to predict the ultimate result of the ceramic materials investigations, but to compute the resulting optimized engine cycle efficiency, vehicle fuel economy, estimated engine manufacturing cost, and resulting consumer cost-of-ownership, assuming such a material proves applicable to the gas turbine engine. Projections of improved component efficiencies and decreased parasitic losses by the year 2000 (based on present rate of technology advancement) are used in this analysis to estimate the ultimate potential of this engine.

The 1985 engine is defined as the engine that could be demonstrated by 1985, assuming that the technology that is expected to exist in the year 2000 is available prior to 1985 at an expenditure of effort considerably higher than is anticipated. The 1995 engine is defined in a similar manner.

The 1985 engine is a single-shaft configuration with a rotary regenerator. The seals problem of the regenerator is expected to be considerably reduced before 1985. This type of heat exchanger provides a smaller engine, lower manufacturing cost, higher fuel economy, and greater turbine-life margin, when compared with a similar engine, using a fixed-boundary recuperator. Although the assumed effectiveness of the 1985 regenerator is higher than that for the 1975 engine (0.94 compared to 0.90, respectively), the diameter of the core is not changed, since this engine has a higher specific power and, therefore, lower airflow rate. As the estimated manufacturing cost of the regenerator core is only $30, there is little incentive to reduce the cost by reducing the regenerator effectiveness.

The anticipated 1995 engine is essentially that of the 1985, except that the variable inlet guide vanes were eliminated to reduce manufacturing cost. Increasing turbine inlet temperature for the 1995 engine tends to improve fuel economy and counteracts the elimination of the inlet guide vanes. Again, the 1995 engine is regenerated rather than recuperated.

12.2 ENGINE PERFORMANCE EVALUATION

Preceding studies showed that regardless of the choice of transmission, the regenerated single-shaft cycle was superior to the free-turbine for exhaust emissions, performance, manufacturing cost, initial consumer cost, cost of ownership, and weight. Therefore, potential advancements of gas turbine, long-range technology were applied to this basic engine cycle. Two engines were studied, and each was compared to the corresponding engine in the previous study. Table 12-1 shows the thermodynamic cycle assumption comparison. Tables 12-2 and 12-3 show the computed sea-level, standard-day, design-point performance for each engine. Each engine was designed to fulfill the Vehicle Design Goals - Six Passenger Automobile constraints adhered to in the preceding studies. The following paragraphs briefly describe the technical approach for the development of design-point and off-design performance for each.

TABLE 12-1

TIME PERIOD COMPARISON OF TECHNOLOGY LEVELS
(Sea-Level, Standard Day)
(Single-Shaft Regenerated Cycle)

Anticipated Engine, year	Pressure Ratio	Turbine Inlet Temp., °F	Variable IGV	Turbine Cooling Air, %	Regenerator		η_c, %	η_t, %	Combustor Pressure Loss, %	Specific Power hp/lb/sec	Specific Fuel Consumption, lb/hp-hr
					Effectiveness, ε_R, %	Leakage, %					
1975	4.6:1	1900	Yes	1.5	90	5.4	82.2	89.0	4	88.6	0.410
1985	4.7:1	2200	Yes	None	94	2.7	85.5	93.0	3	145.9	0.298
1995	5:1	2400	No	None	94	2.7	86	93	3	171.82	0.286

TABLE 12-2

1985 REGENERATED SINGLE-SHAFT CYCLE DESIGN POINT

LHV	H/C	ALTITUDE	FPS
18300.0	0.16786	0.0	0.000

COMPRESSOR			
DRIVING TURBINE	HP	ENERGY CORR. COOLING AIR	WATER
1	86.70	0.000	0.000

TURBINE					
				BEFORE MIXING	
SHAFT HP	ACC HP	HP REQUIRED	MECHANICAL EFFICIENCY	ENTHALPY	TEMP
110.6	6.0	205.7	0.980	508.17	1985.4

BURNER	WATER	FUEL FLOW
	0.000	33.01

	CR FLOW	PRESSURE	TEMP	DELTA	THETA	R	ENTHALPY	GAMMA	F/A	W/A	EFF	P/P	DEL P	LEAKAGE	COOLING
AMBIENT	0.758	14.696	518.7	1.000	1.000	53.349	124.00	1.401	--	0.0000	--	--	--	--	--
INLET	0.769	14.476	518.7	0.985	1.000	53.349	124.00	1.401	--	0.0000	1.000	1.000	0.015	0.000	--
COMPRESSR	0.209	68.035	853.2	4.630	1.645	53.349	204.86	1.389	--	0.0000	0.855	4.700	0.000	0.004	0.000
REGNR CLD	0.316	65.790	1917.4	4.477	3.697	53.349	481.89	1.331	--	0.0000	0.940	1.000	0.033	0.014	--
BURNER	0.388	63.816	2659.7	4.342	5.128	53.370	703.61	1.302	0.0125	0.0000	0.999	1.000	0.030	0.000	--
TURBINE	1.285	16.640	1985.4	1.132	3.828	53.370	508.17	1.320	0.0125	0.0000	0.930	3.835	0.000	0.000	0.000
REGRN HOT	1.041	14.693	961.7	1.000	1.854	53.370	233.75	1.375	0.0121	0.0000	--	1.000	0.117	0.014	--

FUEL FLOW	TOTAL	CORRECTED	SFC
	33.0	33.5	0.298

POWER	NET	CORRECTED	SPECIFIC
	110.6	112.3	145.94

TABLE 12-3

1995 REGENERATED SINGLE-SHAFT CYCLE DESIGN POINT

LHV	H/C	ALTITUDE	FPS
18300.0	0.16786	0.0	0.000

COMPRESSOR			
DRIVING TURBINE	HP	ENERGY CORR. COOLING AIR	WATER
1	76.90	0.000	0.000

TURBINE					
SHAFT HP	ACC HP	HP REQUIRED	MECHANICAL EFFICIENCY	BEFORE MIXING ENTHALPY	BEFORE MIXING TEMP
110.6	6.0	195.9	0.980	545.88	2114.3

BURNER	WATER	FUEL FLOW
	0.000	31.60

	CR FLOW	PRESSURE	TEMP	DELTA	THETA	R	ENTHALPY	GAMMA	F/A	W/A	EFF	P/P	DEL P	LEAKAGE	COOLING
AMBIENT	0.644	14.696	518.7	1.000	1.000	53.349	124.00	1.401	--	0.0000	--	--	--	--	--
INLET	0.653	14.476	518.7	0.985	1.000	53.349	124.00	1.401	--	0.0000	1.000	1.000	0.015	0.000	--
COMPRESSR	0.168	72.378	867.9	4.925	1.673	53.349	208.44	1.388	--	0.0000	0.860	5.000	0.000	0.004	0.000
REGNR CLD	0.260	69.989	2039.4	4.762	3.932	53.349	515.66	1.327	--	0.0000	0.940	1.000	0.033	0.014	--
BURNER	0.322	67.890	2859.7	4.620	5.513	53.373	764.69	1.297	0.0141	0.0000	0.999	1.000	0.030	0.000	--
TURBINE	1.128	16.638	2114.3	1.132	4.076	53.373	545.88	1.314	0.0141	0.0000	0.930	4.080	0.000	0.000	0.000
REGRN HOT	0.900	14.691	993.3	1.000	1.915	53.372	242.00	1.372	0.0137	0.0000	--	1.000	0.117	0.014	--

FUEL FLOW	TOTAL	CORRECTED	SFC
	31.6	32.1	0.286

POWER	NET	CORRECTED	SPECIFIC
	110.6	112.3	171.82

The first task was to estimate the level of compressor and turbine efficiencies potentially possible, by considering unlimited availability of R&D funds. Figure 12-1 shows, as a function of time, a band of compressor polytropic efficiencies comprising the probable limits of centrifugal compressor advancement for large and small airflow rates, assuming the present rate of advancement. The small stage (or polytropic) efficiency describes the compression process locally, at a point in the compressor. When the polytropic efficiency is constant during compression, it is related to the adiabatic efficiency by

$$\eta_{ad} = \frac{\left(\frac{P_{t_2}}{P_{t_1}}\right)^{(\gamma-1)/\gamma} - 1}{\left(\frac{P_{t_2}}{P_{t_1}}\right)^{(\gamma-1)/(\gamma\eta_p)} - 1} = \frac{\text{ideal compression work}}{\text{actual compression work}}$$

where

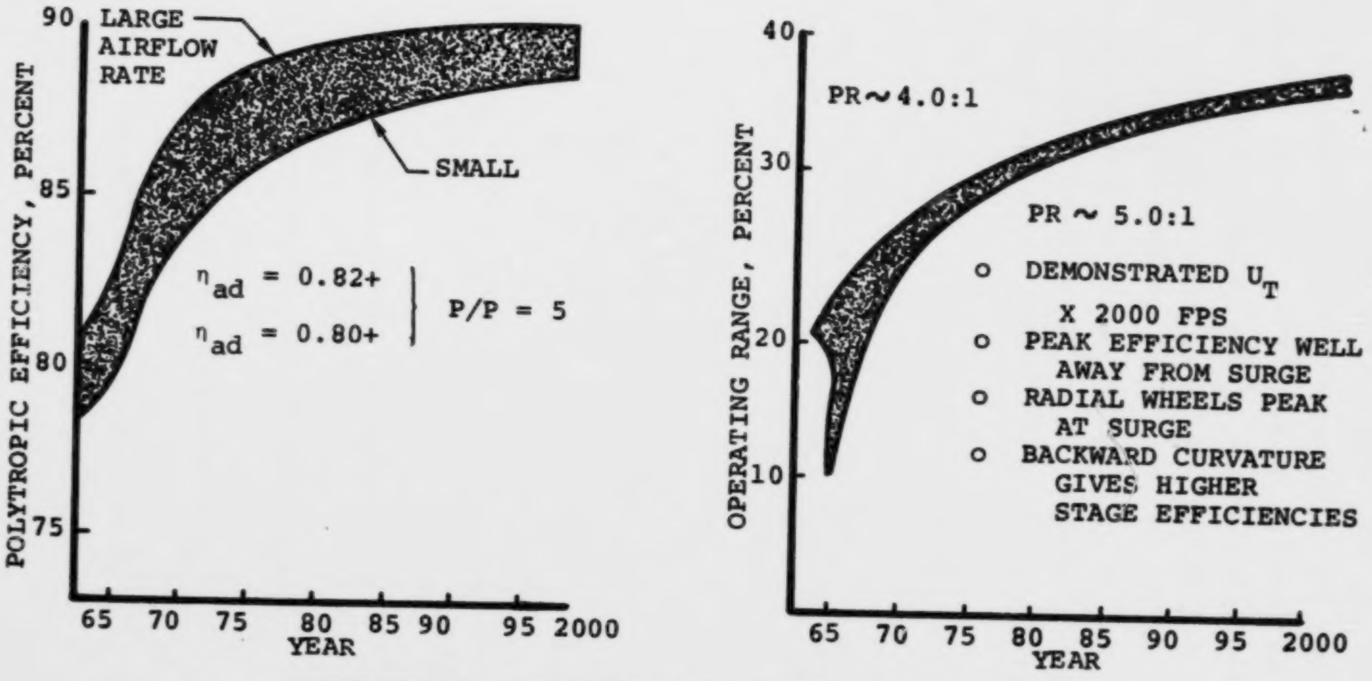
$\frac{P_{t_2}}{P_{t_1}}$ = cycle pressure ratio

γ = ratio of specific heats

η_p = polytropic efficiency

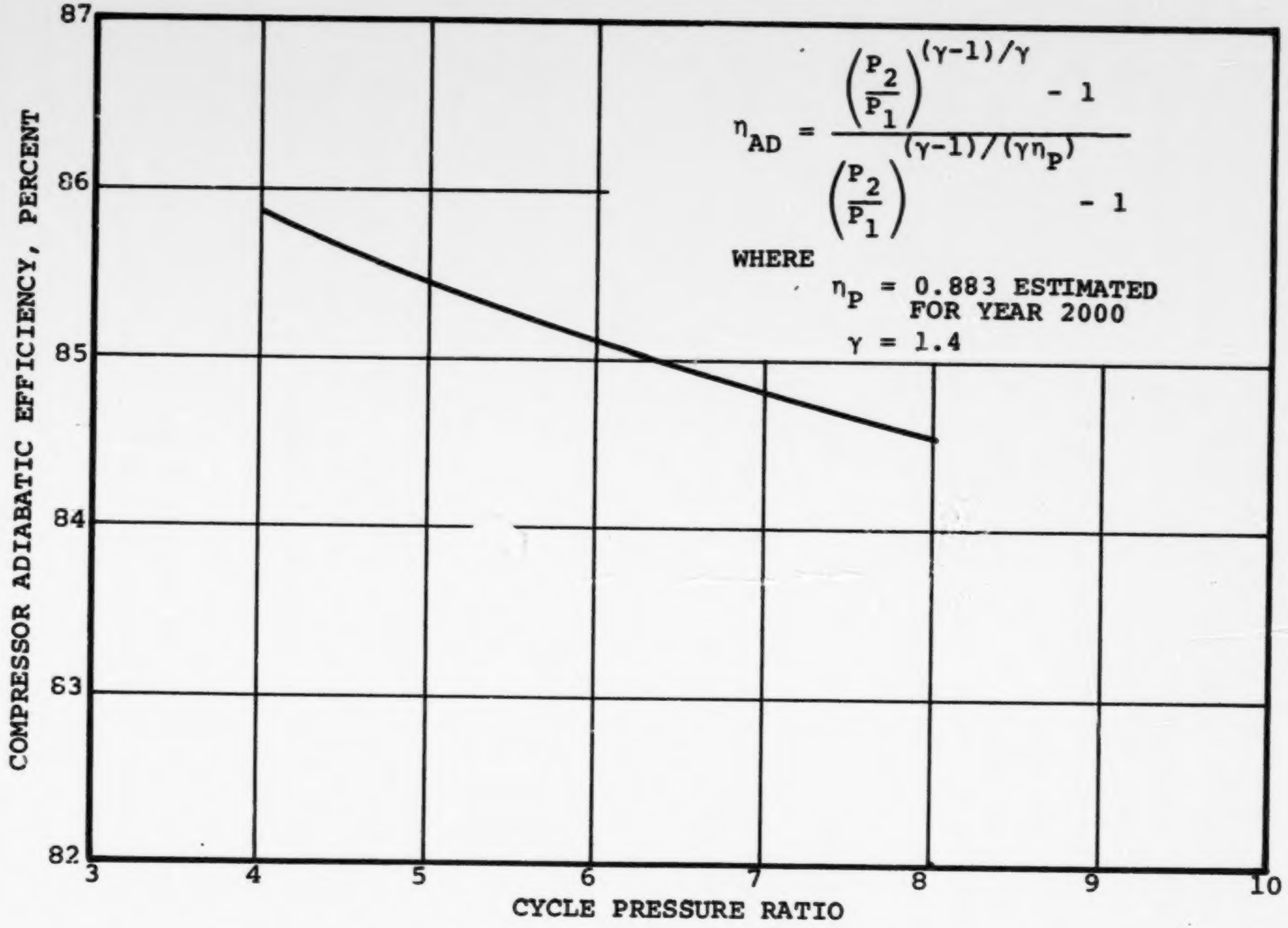
η_{ad} = adiabatic efficiency

Figure 12-1 indicates that a small compressor polytropic efficiency of 0.883 is potentially obtainable by the year 2000. Using this value, a curve of compressor adiabatic efficiency was calculated as a function of cycle pressure ratio (Figure 12-2).



CENTRIFUGAL COMPRESSOR PERFORMANCE ADVANCEMENT

FIGURE 12-1



$$\eta_{AD} = \frac{\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1}{\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/(\gamma\eta_P)} - 1}$$

WHERE

η_P = 0.883 ESTIMATED FOR YEAR 2000

$\gamma = 1.4$

PREDICTED COMPRESSOR EFFICIENCY
WITH PRESSURE RATIO FOR YEAR 2000

FIGURE 12-2

Figure 12-3 shows a plot of radial inflow turbine adiabatic efficiency (total-to-total) versus time over a range of pressure ratios from 3 to 5 and for corrected airflows in the range of interest. This curve was entered at the year 2000, and turbine efficiency was obtained for cycle pressure ratios of 3 and 5. A straight-line approximation of efficiency variation between these pressure ratio limits was compiled for Figure 12-4. Additional thermodynamic assumptions were made for the parametric design-point studies. These include:

(a) Zero turbine cooling flow

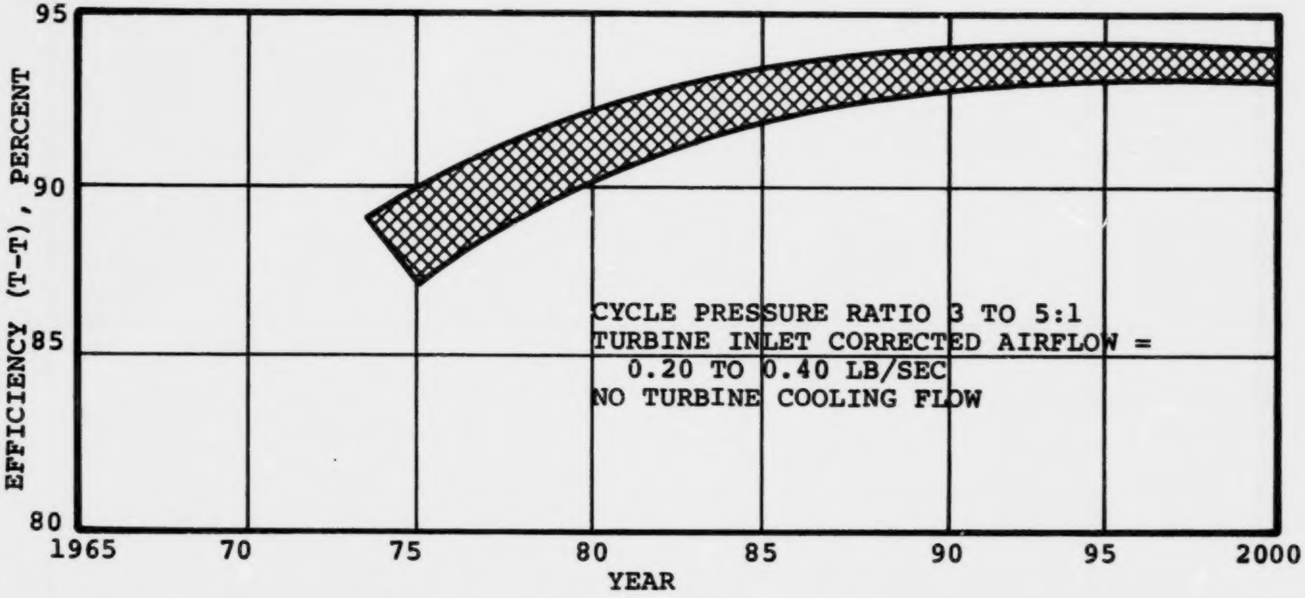
(b) Regenerator leakage flow of one-half of the values used in the preceding studies for a given cycle pressure ratio

(c) Combustor pressure drop equal to 3 percent

(d) All other cycle pressure losses as in the preceding study

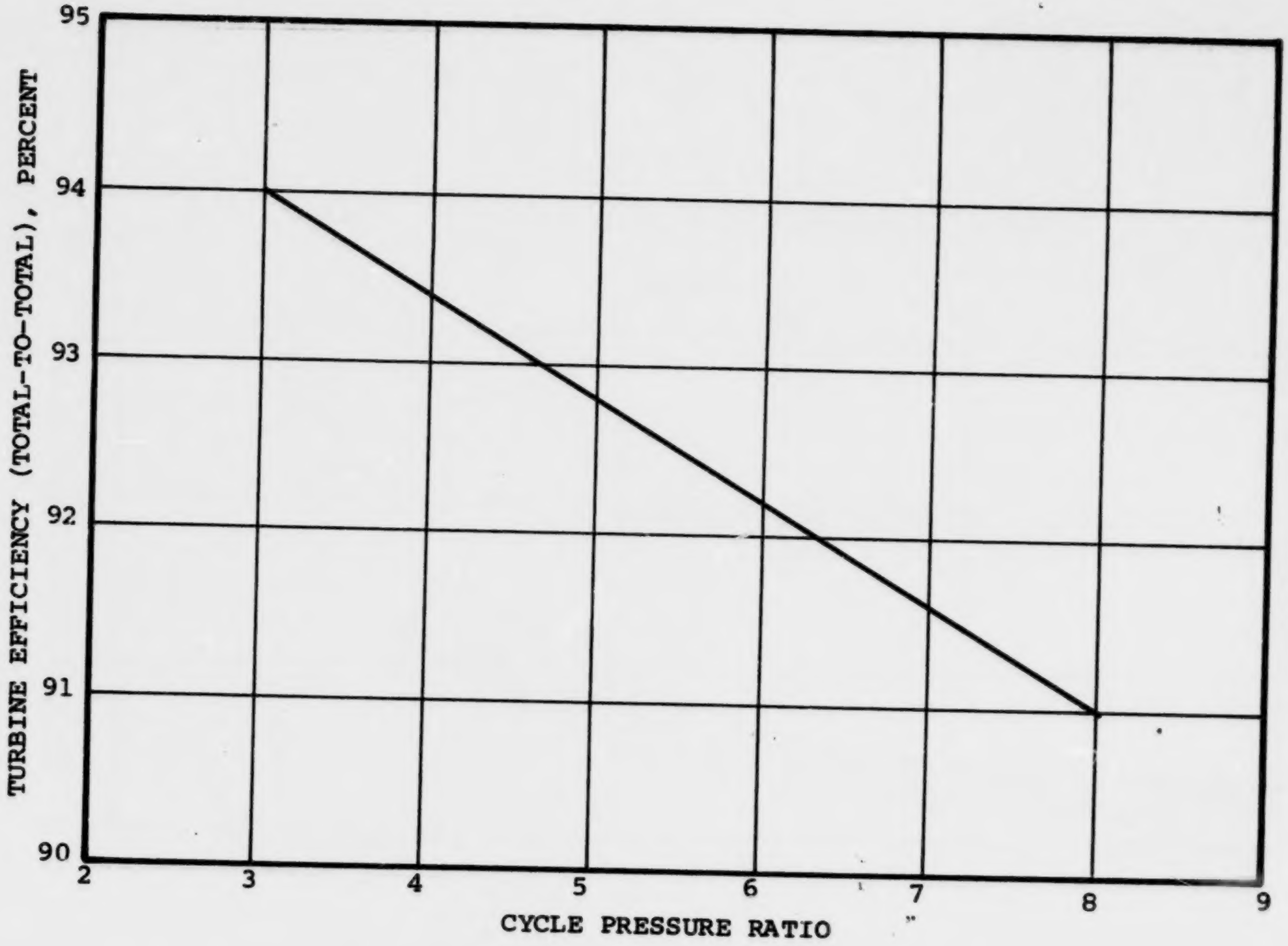
Figure 12-5 shows one of the plots resulting from the parametric cycle studies. Specific power and specific fuel consumption are plotted for a range of cycle pressure ratios from 4 to 8 and turbine inlet temperatures from 1800° to 2400°F for a constant regenerator effectiveness of 0.94. As indicated on Tables 12-1, 12-2, and 12-3, a pressure ratio of 4.7:1 and a turbine inlet temperature of 2200°F were selected for the 1985 engine. A pressure ratio of 5:1 and a turbine inlet temperature of 2400°F were selected for the 1995 engine. Table 12-4 summarizes the design-point parameters for each engine.

Since the vehicle acceleration goals must be met on an 85°F day at sea level, off-design (or part-load) engine performance maps were generated accordingly (Figures 12-6 and 12-7). Each engine was scaled to a rating of 110.6 shp in the mission analysis program, in order to attain the vehicle performance goals. Compressor inlet guide vanes



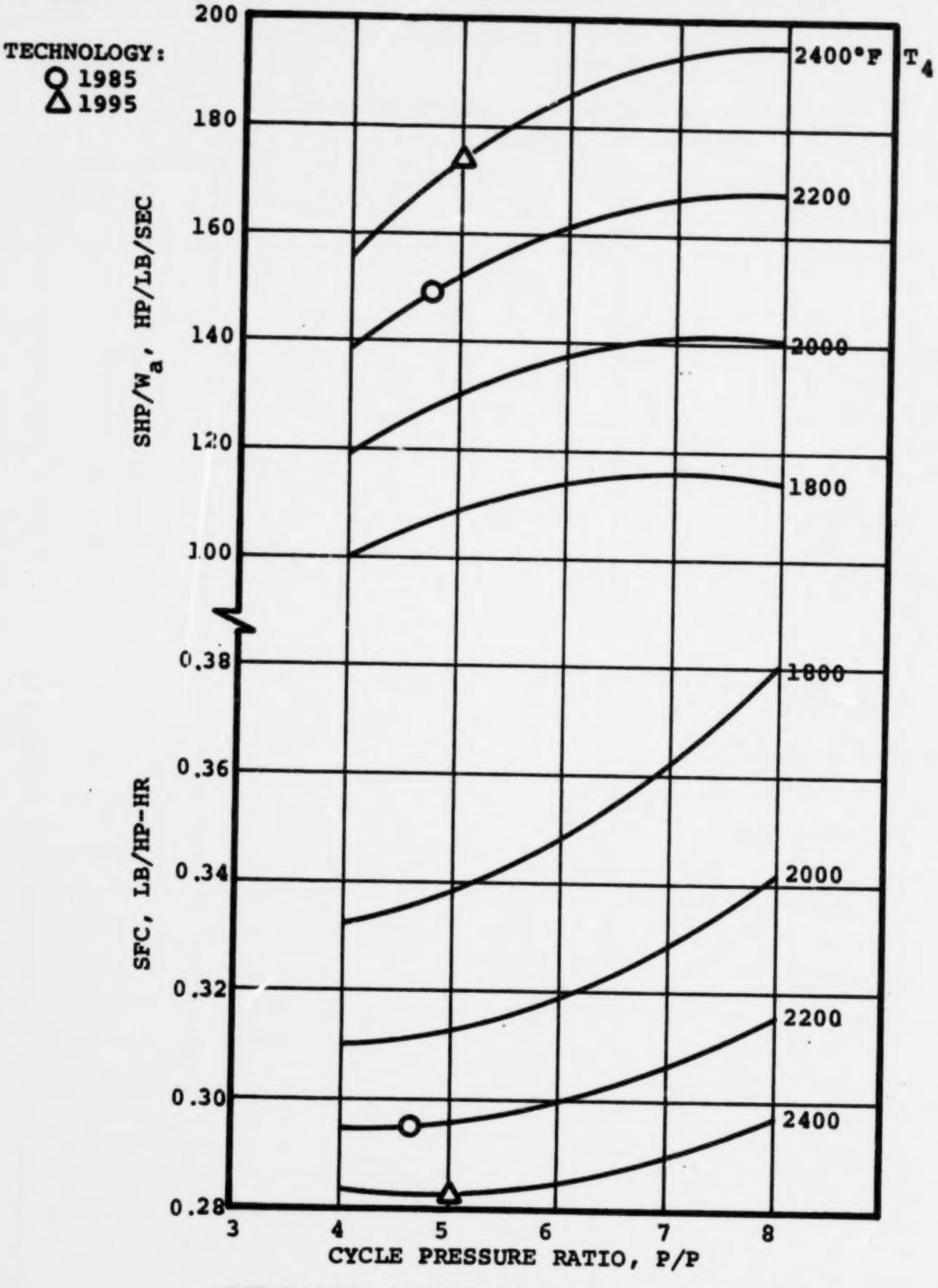
RADIAL IN-FLOW TURBINE PERFORMANCE ADVANCEMENT

FIGURE 12-3



PREDICTED RADIAL TURBINE EFFICIENCY VARIATION
WITH PRESSURE RATIO FOR YEAR 2000

FIGURE 12-4



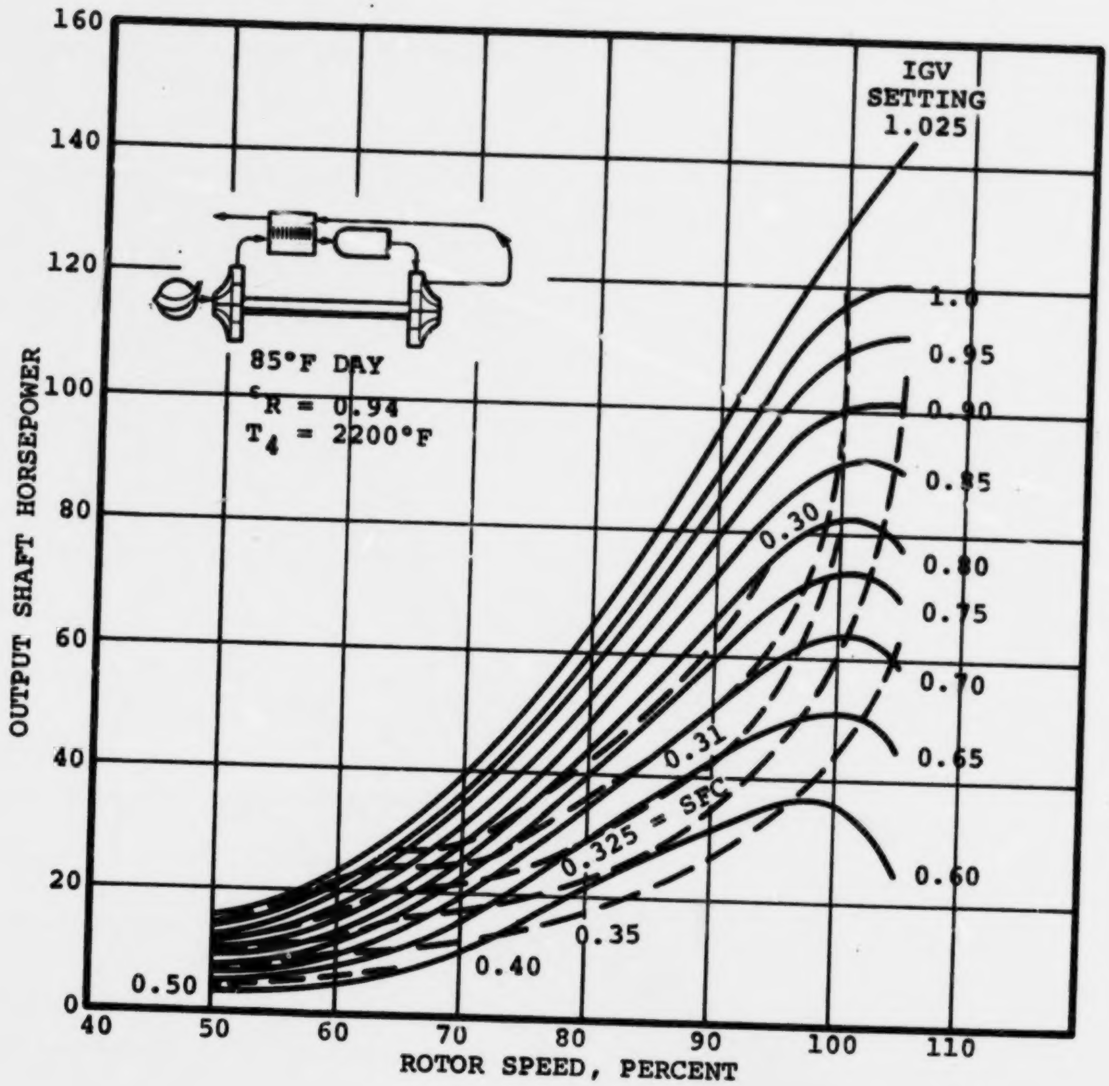
REGENERATED SINGLE-SHAFT CYCLE (ε_R = 0.94)

FIGURE 12-5

TABLE 12-4

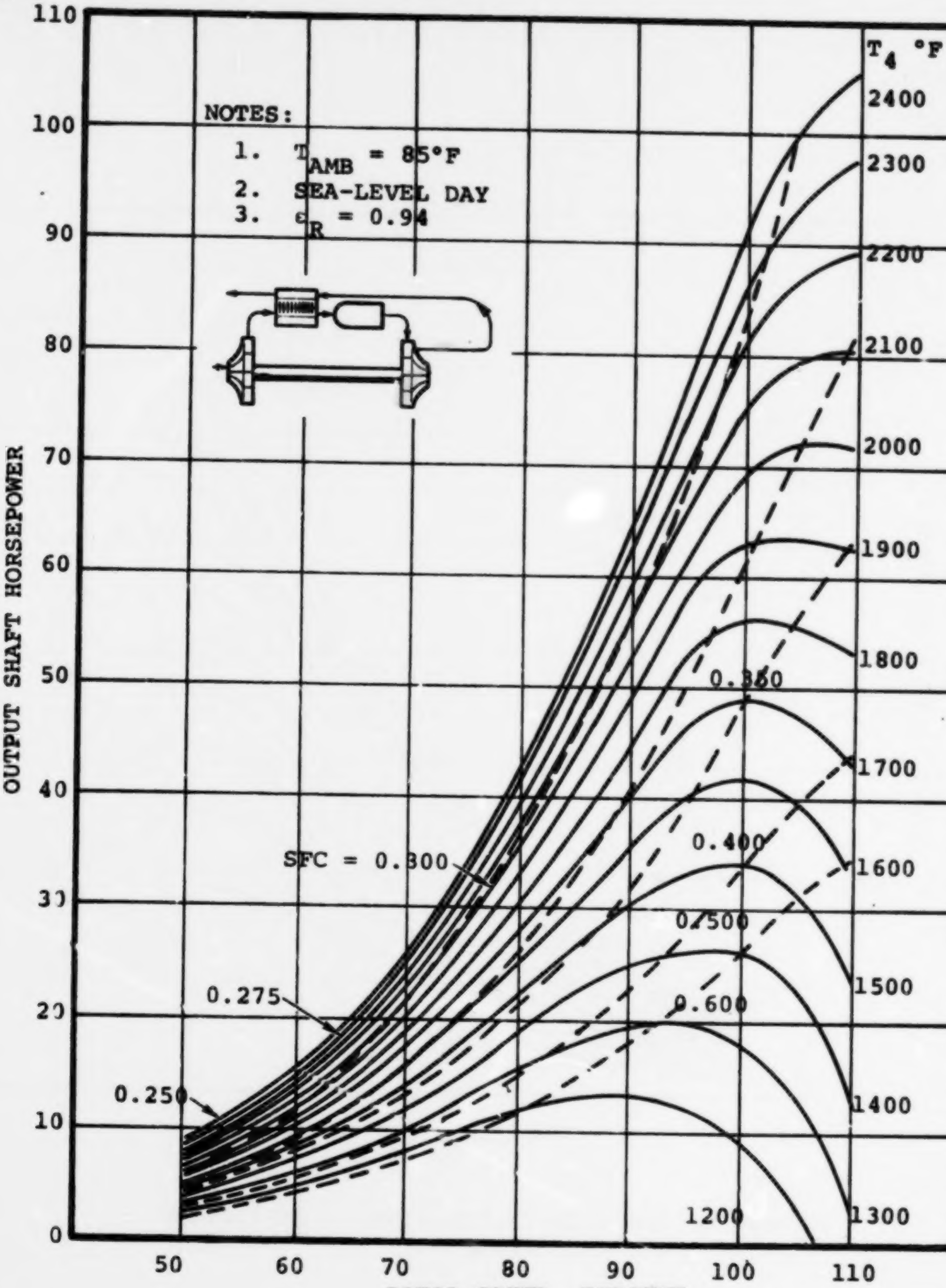
DESIGN-POINT PARAMETERS AND PERFORMANCE (Sea-Level, Standard Day)

Specifications	Unit	1985	1995
Compressor			
Inlet temperature	°R	518.7	518.7
Airflow	lb/sec	0.758	0.644
Pressure ratio	P/P	4.7:1	5.0:1
Efficiency	%	85.5	86.0
Speed	rpm	107,000	115,500
Impeller diameter	in.	3.62	3.38
Polar moment of inertia	lb-in.-sec^2	0.00062	0.00041
Materials		Al	Al
Combustor			
Inlet temperature	°R	1917.4	2039.4
Efficiency	%	99.9	99.9
Radial Inflow Turbine			
Inlet temperature	°F	2200	2400
Pressure ratio	P/P	3.835:1	4.080:1
Corrected inlet airflow	lb/sec	0.388	0.322
Efficiency	%	0.93	0.93
Rotor diameter	in.	4.0	3.74
Polar moment of inertia	lb-in.-sec^2	0.0068	0.0045
Material		Ceramic	Ceramic
Heat Exchanger		Rotary Generator	
Effectiveness		0.94	0.94
Leakage	%	2.7	2.7
Material		Ceramic	Ceramic
Performance			
Specific fuel consumption	lb/hp-hr	0.298	0.286
Power	shp	110.6	110.6



1985 ENGINE REGENERATED SINGLE-SHAFT CYCLE
SEA-LEVEL 85°F DAY

FIGURE 12-6



1995 ENGINE SINGLE-SHAFT REGENERATED

FIGURE 12-7

were used to obtain the performance map for the 1985 engine (Figure 12-6). These vanes were required to improve the specific fuel consumption and achieved their purposes by permitting the engine to operate at a constant turbine inlet temperature of 2200°F at part-load. The engine performance is for guide vane settings yielding flow ratios of 0.6 to 1.025.

No inlet guide vanes were used for the 1995 engine, since the increased cost attributed to the required controls and actuation device could not be justified. The increased cost was weighed against the benefit from the higher turbine inlet temperatures and resultant simplicity of the engine design and was abandoned.

The idle speed for both engines was selected as 60 percent of design-rated speed. This selection was a compromise between fuel consumption and engine acceleration time.

12.3 VEHICLE PERFORMANCE AND FUEL ECONOMY

Anticipated vehicle performances with the 1985 and 1995 engines are compared to the 1975 engine on Table 12-5 for an 85°F day. The idle-to-maximum torque acceleration times for the 1985 and 1995 engines are slightly lower than that for the 1975 engine, as a result of the smaller engine inertias for the advanced designs. The reason for the slightly higher elapsed times for the advanced engines (the 0 to 60 and 25 to 70 mph, and the DOT pass maneuvers) is that the overspeeds of these engines were limited to 104 percent of design speed. The 1975 engine could run at 109 percent of design speed, allowing about a 5 percent higher peak power. The off-design performance maps for the advanced engines were not computed for a 105°F day. However, based on the above similarity of engines, the vehicle acceleration performance on a 105°F day for the advanced engines will be essentially the same as that given on Table 7-3 for the 1975 engine.

TABLE 12-5

PERFORMANCE COMPARISONS
(4000-lb Automobile, 85°F Day)

Anticipated Engine,[1] year	Power[2] Boost, %	Acceleration Times					Grade Velocity, mph		Idle Fuel Flow, Percent of Maximum	Rated hp, 59°F Day
		Idle-to-Maximum Output Torque, sec	0 to 440 ft sec	0 to 60 mph sec	25 to 70 mph sec	DOT Pass sec	5%	30%[3]		
1975	24	0.9	9.9	11.3	12.5	14.5	90	25+	5.6	108
1885	↓	0.8	↓	11.9	13.5	14.2	↓	↓	4.8	↓
1995	↓	0.8	↓	11.8	13.4	14.2	↓	↓	7.2	↓

[1] Belt transmission

[2] With water injection; water flow rate = approximately 4 to 6 percent of air mass flow rate for each engine.

[3] 4700-lb vehicle

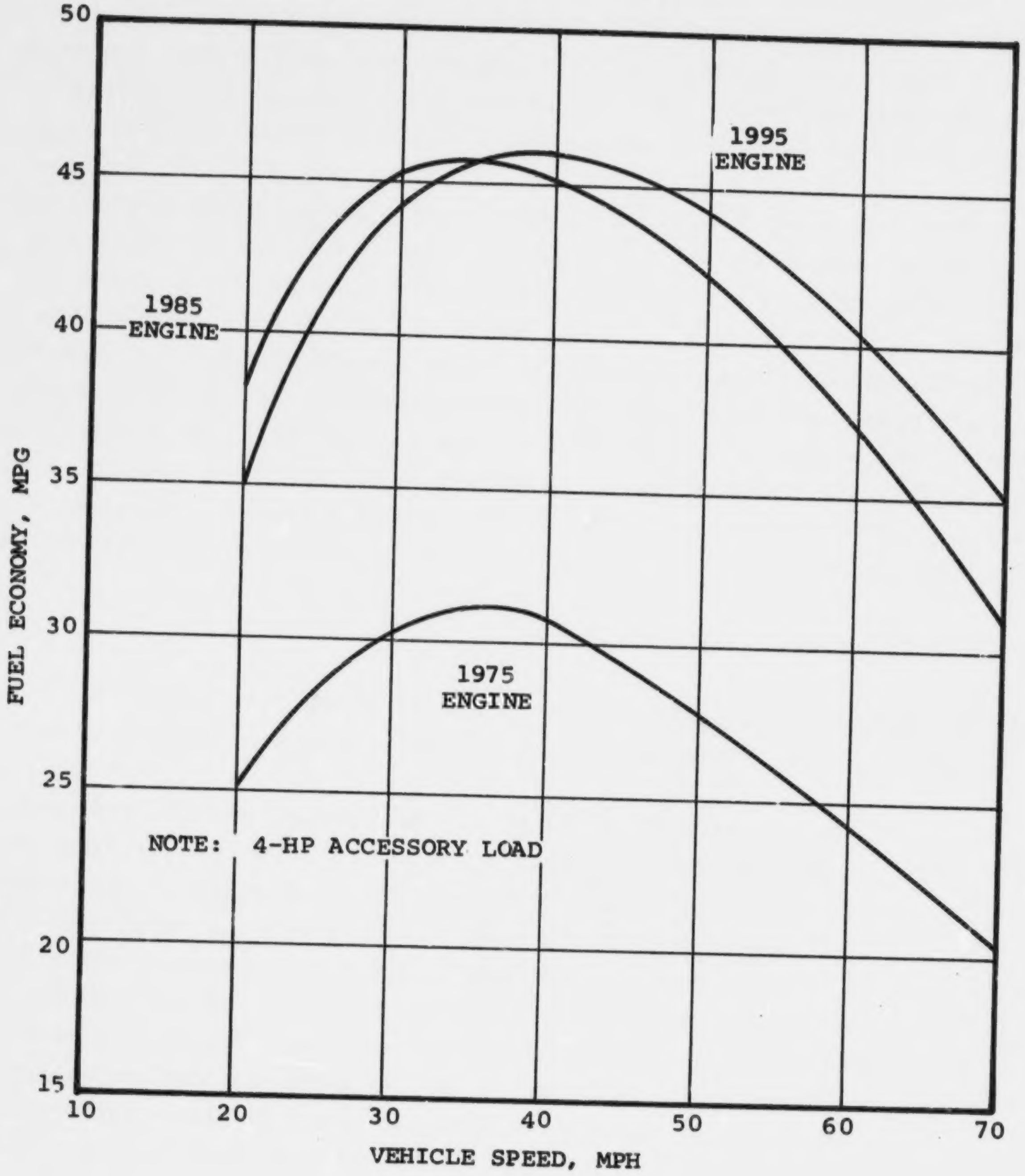
Note that the rated power of the advanced engines is the same as for the 1975, even though the advanced units have lower engine inertias. The reason is that the 25 to 70 mph and the DOT pass maneuvers require that the engine produce approximately 100 hp, even if the engine had zero inertia or operated at constant speed.

Constant speed fuel economy comparisons of the three engine configurations are shown in Figure 12-8. Fuel economy comparisons for the Federal Driving Cycle (FDC) and the Simplified Driving Cycle (USEDC) (Appendix I, Table 1) are shown on Table 12-6. Both of the advanced engines obtain better than twice the fuel economy of the 1970 spark-ignition engine on the FDC and the USEDC. The 1985 engine has better fuel economy than the 1995, since at the lower power operating conditions, the variable inlet guide vanes of the former provide higher overall cycle efficiency. The variable guide vanes were eliminated on the 1995 engine to reduce manufacturing cost and consumer initial cost.

12.4 MANUFACTURING COST ESTIMATES

The following assumptions were made for manufacturing costs of the 1985 and 1995 engines in terms of 1971 dollars and labor rates:

(a) Probably all of the parts in the turbine section and most of the remainder of the engine would have to be redesigned to accept the ceramic turbine. Also, because of the higher specific powers of the 1985 and 1995 engines relative to the 1975 unit, most of the larger and more expensive components will be scaled down as a function of the reduced design-point airflows of the advanced engines. However, present speculation on redesign of engine parts assumes no change in the material cost in dollars per pound or the manufacturing labor cost; the only cost change is that due to weight change in scaling down the engine components.



CONSTANT-SPEED FUEL ECONOMY COMPARISONS

FIGURE 12-8

TABLE 12-6

FUEL ECONOMY COMPARISONS
(4000-lb automobile, 85°F day, accessory load 1.3 hp FDC, 4.0 hp for all other routes)

Engine Configuration, year	Federal Driving Cycle, mpg	Composite USEDC, mpg	Rated hp 59°F Day Without Boost, shp
1970 V-8 Spark Ignition Automatic Transmission	12.5	14.8	175
1975*	17.0	22.1	108
1985*	27.6	34.3	↓
1995*	25.8	33.5	

*Belt transmission

(b) The lower density of the ceramic material will not affect the cost, since the ceramic version of a given part is expected to have a larger volume because of the reduced strength of the ceramic material.

(c) The reduction gearbox for the advanced engines will have a larger overall speed ratio because of the higher engine speeds. The necessary speed reductions will be achieved with the same number of gear meshes, but ratios will be slightly increased for each mesh.

(d) All other manufacturing cost assumptions are the same as those for the 1975 engine, as specified in Section 8.

The estimated manufacturing costs of the 1985 and 1995 engines, relative to the 1975 version, are shown on Tables 12-7 and 12-8. The manufacturing cost of the 1985 engine is estimated as approximately $349 or about $80 less than the 1975; similarly, the cost of the 1995 engine is estimated at $300.

The total estimated automobile manufacturing and consumer costs with the advanced engines are compared to those of the 1975 gas turbine and the 1970 spark-ignition engines on Table 12-9. The consumer cost of the 1995 vehicle is estimated as about $300 more than that of the 1970 (Appendixes 4 and 5).

Weights for a standard-size, six-passenger automobile with four engine-transmission systems are summarized on Table 12-10. The reduction in weight for the 1985 and 1995 versions is mostly the result of smaller engine components. The control system actuator for the variable IGVs is eliminated on the 1995 engine, thus reducing the control system weight by 8 lb.

TABLE 12-7

1985 ENGINE

Find Number	Description	Finished Weight, lb	Material Cost, $	Δ Cost, $	Δ Weight, lb
1	Impeller	0.29	1.60	- 1.70	- 0.31
10	Regenerator Core*	17.27	30.00	-	+ 6.27
	Turbine				
2	Ceramic Wheel	1.85	15.70	-19.30	- 2.26
13	Scroll	3.88	14.20	-15.80	- 4.29
14	Shroud	3.62	8.50	- 9.50	- 4.02
	Housing				
16	Inlet	10.20	7.15	- 7.85	-11.21
19	Diffuser	9.00	8.15	- 5.45	- 6.00
42	Center	36.6	31.20	-20.80	-24.40
TOTAL				-80.40	-46.22

*No cost change; approximately same diameter as 1975 engine

Notes:

1. Find numbers refer to Figure 8-5
2. Δ Cost = Cost for 1985 engine - cost for 1975 engine
3. Δ Weight = Weight for 1985 engine - weight for 1975 engine

TABLE 12-8

1995 ENGINE

Find Number	Description	Finished Weight, lb	Material Cost, $	Δ Cost, $	Δ Weight, lb
1	Impeller	0.21	1.15	- 3.65	- 0.39
10	Regenerator Core*	13.50	30.00	-	+ 2.50
	Turbine				
2	Ceramic Wheel	1.41	12.00	- 23.00	- 2.70
13	Scroll	2.80	10.30	- 19.70	- 5.37
14	Shroud	2.62	6.20	- 11.80	- 5.02
	Housing				
16	Inlet	7.35	5.15	- 9.85	-14.06
19	Diffuser	7.35	6.70	- 6.90	- 7.65
42	Center	30.00	25.50	- 26.50	-31.00
55-62**	Miscellaneous	0	0	- 27.26	- 3.60
TOTAL				-128.66	-67.29

*No change in cost; diameter is approximately same as 1975 engine.

**IGVs eliminated

Notes:

1. Find numbers refer to Figure 8-5
2. Δ Cost = Cost for 1995 engine - cost for 1975 engine
3. Δ Weight = weight for 1995 engine - weight for 1975 engine

TABLE 12-9

SUMMARY OF ESTIMATED AUTOMOBILE MANUFACTURING AND CONSUMER COSTS, DOLLARS
(Full-size, six-passenger, four-door sedan)

Engine Configuration, year	Vehicle Components				All Other Parts	Total Costs	
	Engine	System Controls	Accessories	Transmission		Manufacturing	Consumer
1970 V-8 Spark Ignition Automatic Transmission	356[1]	-	-	267	1157[2]	1780	3185
1975[3]	429	137	61	197	1150	1974	3723
1985[3]	349	137	↓	↓	↓	1894	3576
1995[3]	300	116	↓	↓	↓	1824	3456

[1] Includes controls and engine-mounted accessories

[2] Seven-pound battery included in accessories

[3] Belt transmission

TABLE 12-10

VEHICLE WEIGHT SUMMARY, POUNDS
(Full-Size, Six-Passenger)

Engine Configuration, year	Engine	Transmission	Hydromechanical Controls	Accessories	Exhaust Ducting	All Other Parts	Total
1970 V-8 Spark Ignition Automatic Transmission	565[1]	160	-	-	55	3135[3]	3915
1975[2]	256	132	22	68	100	3085	3663
1985[2]	210	↓	22	↓	↓	↓	3617
1995[2]	189	↓	14	↓	↓	↓	3588

[1] Includes control system and accessories

[2] Belt transmission

[3] Includes 50-lb battery; battery weight included with accessories for gas turbine

12.5 COST OF OWNERSHIP

Cost-of-ownership comparisons of automobiles equipped with the 1975, 1985, and 1995 gas turbine engines and the 1970 spark-ignition engine are given on Table 12-11. Depreciation for all vehicles was assumed to be at zero scrap value. The cost of JP-4 fuel was based on $0.35/gal and includes a water cost of $0.01/gal of fuel; leaded 90-96 octane gasoline was at $0.036/gal. In Column 6 of the table, Other Parts, the costs include tires, insurance, taxes, etc., as discussed in Appendix 6.

Both of the advanced engines have a total cost-of-ownership slightly less than $10,000 for the life of the vehicle or a total cost of approximately $2000 less than that for the 1970 automobile with conventional system.

TABLE 12-11

COST-OF-OWNERSHIP COMPARISONS, DOLLARS
(Life of 7 years, 105,000 mi; full-size, six-passenger automobile)

Engine Configuration, year	Depreciation	Repairs and Maintenance	Fuel and Water Cost	Oil Cost	All Other Parts	Total Cost of Ownership	Percent of First System
1970 V-8 Spark Ignition Automatic Toansmission	3185	1522	2560	158	4469	11,894	100
1975*	3723	851	1670	-	↓	10,713	90
1985*	3576	↓	1071	-		9,918	83
1995*	3456		1098	-		9,825	83

*Belt transmission

SECTION 13

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LIST OF CREDITS

During the course of the study reported herein, numerous manufacturers in automotive-related industries were contacted for information within their respective disciplines. The following is a listing of major respondents:

- Borg-Warner, Ltd., Transmission Division, England
- Cartriseal Corporation
- Ford Motor Company
- Gates Rubber Company
- General Motors Corporation, Delco Remy Division
- GITS Bros.
- Mobil Oil Company
- Monsanto Chemical Company
- New Departure - Hyatt Division
- Owens - Illinois
- Rockford Clutch
- SKF Industries
- Stauffer Chemical Company
- Sunstrand Aviation
- TRACOR Incorporated
- Corning Glass Works

SUPPLEMENT

SYSTEMS CONTROLS

SUPPLEMENT

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SUPPLEMENT

SYSTEM CONTROLS

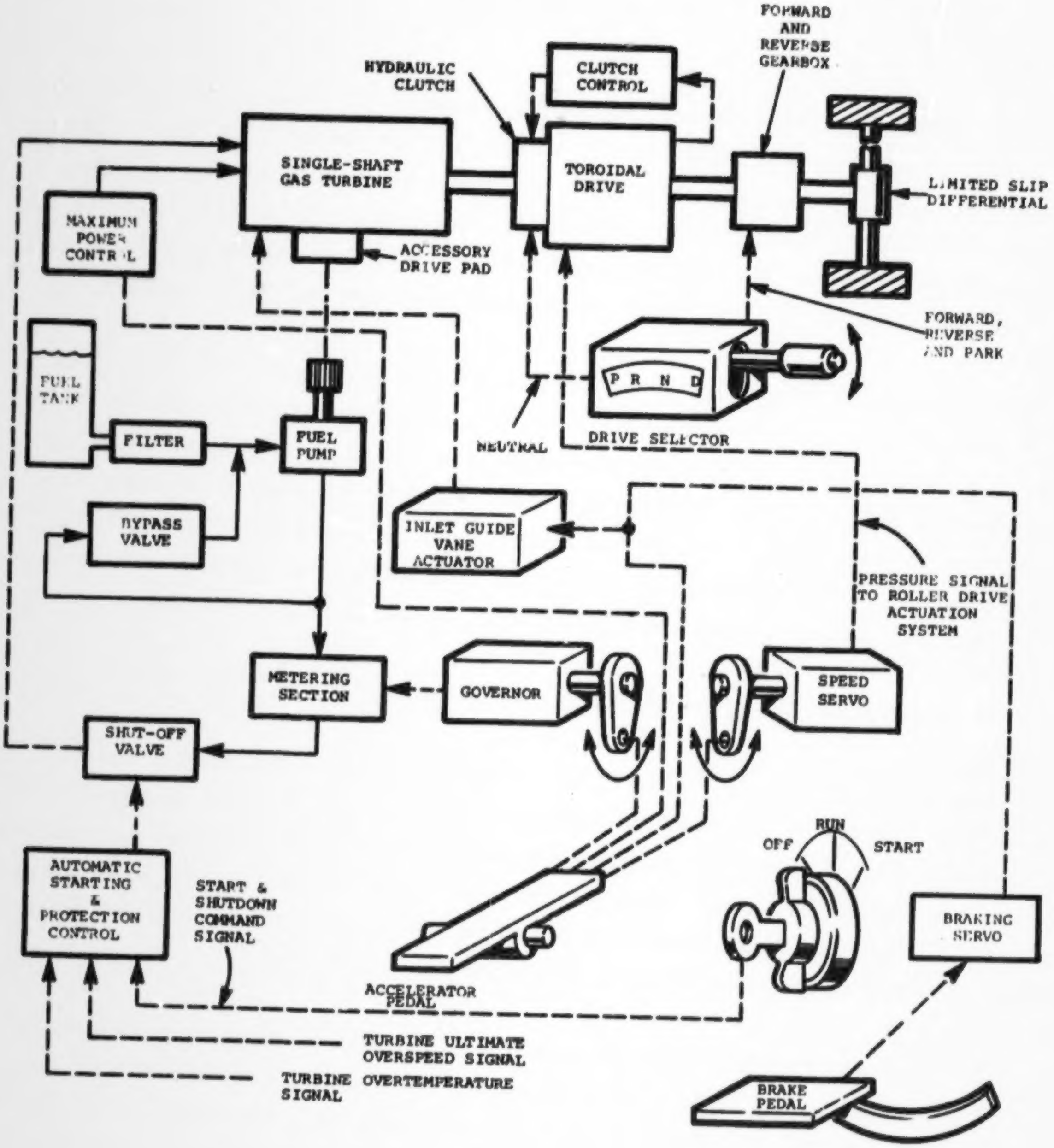
The total systems control concept, essential for controlling a gas-turbine-powered automobile, is described in detail in the following paragraphs for the single-shaft and free-turbine powered vehicles. Total system considerations are particularly useful in determining the unique relationships that exist between the engine, transmission, and fuel-management system to achieve the desired system performance.

In this analysis, the control system included all of the functions and components that are required for optimum operation of the engine. However, in a subsequent analysis of trade-offs between system optimum efficiency and minimum cost to the consumer, it may be desirable to simplify or eliminate parts of the system analyzed in this report. Herein, vehicle cost-of-ownership was emphasized more strongly than initial consumer cost as the optimizing parameter.

1. SINGLE-SHAFT GAS TURBINE CONTROL SYSTEM DESCRIPTION

The control system functional schematic for a variable inlet guide vane, single-shaft, gas-turbine-powered vehicle is shown in Figure 1. The engine is coupled to the rear wheels through a toroidal drive transmission and a forward/reverse-type gearbox. Engine operation with zero vehicle speed is provided by a hydraulic clutch that decouples the drive train from the engine.

Driver control and operation is identical to that for existing automatic transmission vehicles powered by a spark-ignition engine. As a result, standard items such as a drive gear selector, a foot-operated accelerator and brake pedal, and an ignition switch (with off, run, and start positions) are used to control the vehicle.


FORWARD AND REVERSE GEARBOX
HYDRAULIC CLUTCH
CLUTCH CONTROL
SINGLE-SHAFT GAS TURBINE
TOROIDAL DRIVE
LIMITED SLIP DIFFERENTIAL
MAXIMUM POWER CONTROL
ACCESSORY DRIVE PAD
FORWARD, REVERSE AND PARK
P R N D
FUEL TANK
FILTER
FUEL PUMP
NEUTRAL
DRIVE SELECTOR
BYPASS VALVE
INLET GUIDE VANE ACTUATOR
PRESSURE SIGNAL TO ROLLER DRIVE ACTUATION SYSTEM
METERING SECTION
GOVERNOR
SPEED SERVO
SHUT-OFF VALVE
OFF
RUN
START
AUTOMATIC STARTING & PROTECTION CONTROL
START & SHUTDOWN COMMAND SIGNAL
BRAKING SERVO
ACCELERATOR PEDAL
TURBINE ULTIMATE OVERSPEED SIGNAL
TURBINE OVERTEMPERATURE SIGNAL
BRAKE PEDAL

CONTROL SYSTEM FUNCTIONAL SCHEMATIC, SINGLE-SHAFT ENGINE

FIGURE 1

Essential controlling elements in this system include the engine fuel management components and transmission control components. Although somewhat more complex, fuel management of a gas turbine can be compared in function to a carburetor system of a spark-ignition engine, whereas transmission control shows little, if any, deviation from current useage.

Nonessential controlling elements include items such as the inlet guide vane actuator (utilized for part-throttle fuel economy) and the power boost control for maximum acceleration. These systems are included because of their utility in reducing the consumer cost-of-ownership.

Total systems considerations are especially useful in determining the unique relationships that exist between the engine, transmission, and fuel management system to achieve the desired system performance.

Seven major elements comprise the control system:

(a) Fuel control (for fuel management)

(b) Inlet guide vane actuator

(c) Automatic starting and protection control

(d) Transmission control

(e) Hydraulic clutch control

(f) Power boost control

(g) Sensors

Several implementation schemes for employing these elements were studied and are presented, with trade-off factors, in the text that follows. When reference is made to a control type, such as hydromechanical or electronic, the implication is that the computing sections of the control are one of these.

1.1 FUEL CONTROL SYSTEMS AND DESIGN

The primary objectives of controlling fuel to the gas turbine were to:

(a) Accelerate and decelerate as rapidly as possible

(b) Prevent compressor surge

(c) Limit turbine temperature

(d) Prevent burner blowout

(e) Maintain a desired speed set-point under engine load variations

These items are essential to all gas turbine engines for smooth and reliable performance. In addition, compressor or turbine variable guide vanes could be utilized to increase engine efficiency and aid in acceleration and braking control.

Fuel-control component design was essentially determined by scheduling and on-speed fuel requirements of the engine. Scheduling involved the consideration of acceleration time, compressor surge, turbine temperature, and burner blowout, whereas on-speed control involved considerations of turbine temperature, burner blowout, and speed set-point. Typical parameters available for determining fuel control component design were:

(a) Fuel scheduling for:

   (1) Compressor inlet temperature, T_2

   (2) Turbine inlet temperature, T_4

   (3) Turbine exhaust temperature, T_5

   (4) Compressor discharge pressure, P_3

(5) Engine speed, N_E

(6) Corrected engine speed, $N_E / \sqrt{T_2}$

(7) Time, t

(b) On-speed fuel control for:

(1) Engine speed, N_E

(2) Corrected engine speed $N_E / \sqrt{T_2}$

When variable guide vanes were used, component design was mainly determined by the required speed of response and the required steady-state accuracy. Typical parameters available, for determining variable guide vane component design, were:

(a) Steady-state parameter:

(1) Compressor inlet temperature, T_2

(2) Turbine inlet temperature, T_5

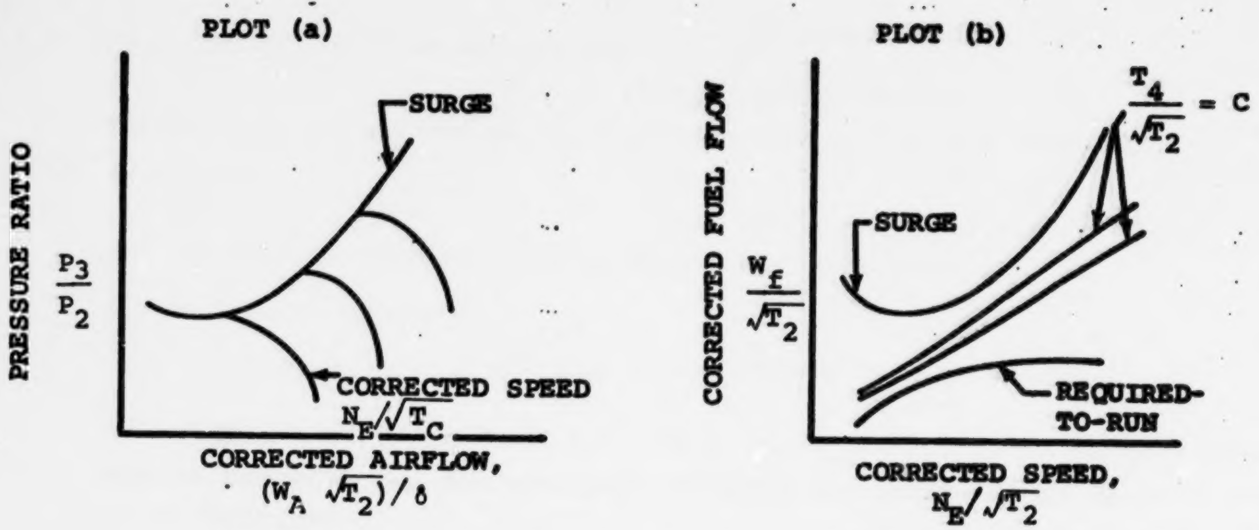
(3) Engine speed, N_E

(4) Corrected engine speed, $N_E / \sqrt{T_2}$

(b) Transient (acceleration and braking) - Engine speed demand, N_D

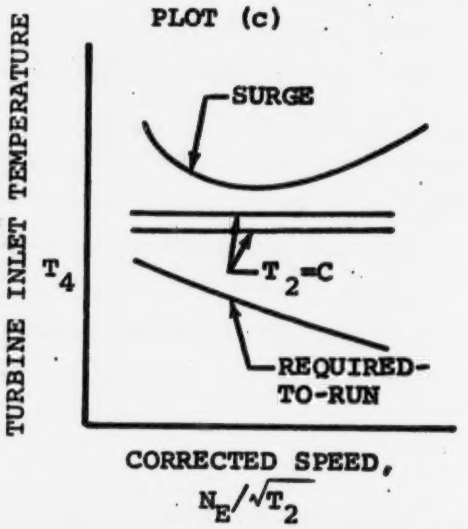
1.1.1 Implementation

Fuel system control implementation was primarily a function of cost and reliability. Cost was a relative quantity and depended largely on required complexity. To determine control complexity, a very thorough study of engine operation was required. For control design purposes, engine operation was normally presented in graphical form as shown on the three plots of Figure 2.



COMPRESSOR MAP

CORRECTED FUEL FLOW
VERSUS
CORRECTED ENGINE SPEED



TURBINE INLET TEMPERATURE
VERSUS
CORRECTED SPEED

ENGINE DATA USED TO DETERMINE CONTROL IMPLEMENTATION

FIGURE 2

Plot (a) defines the compressor characteristics of a gas turbine; Plot (b) defines the bounds on engine fuel flow and speed; and Plot (c) delineates the bounds on engine turbine inlet temperature and speed. These drawings were used to determine fuel scheduling and served to set variable guide vane operation. Fuel control complexity greatly depended on fuel scheduling, as dictated by the shape and separation of the surge and required-to-run lines shown on Plot (b).

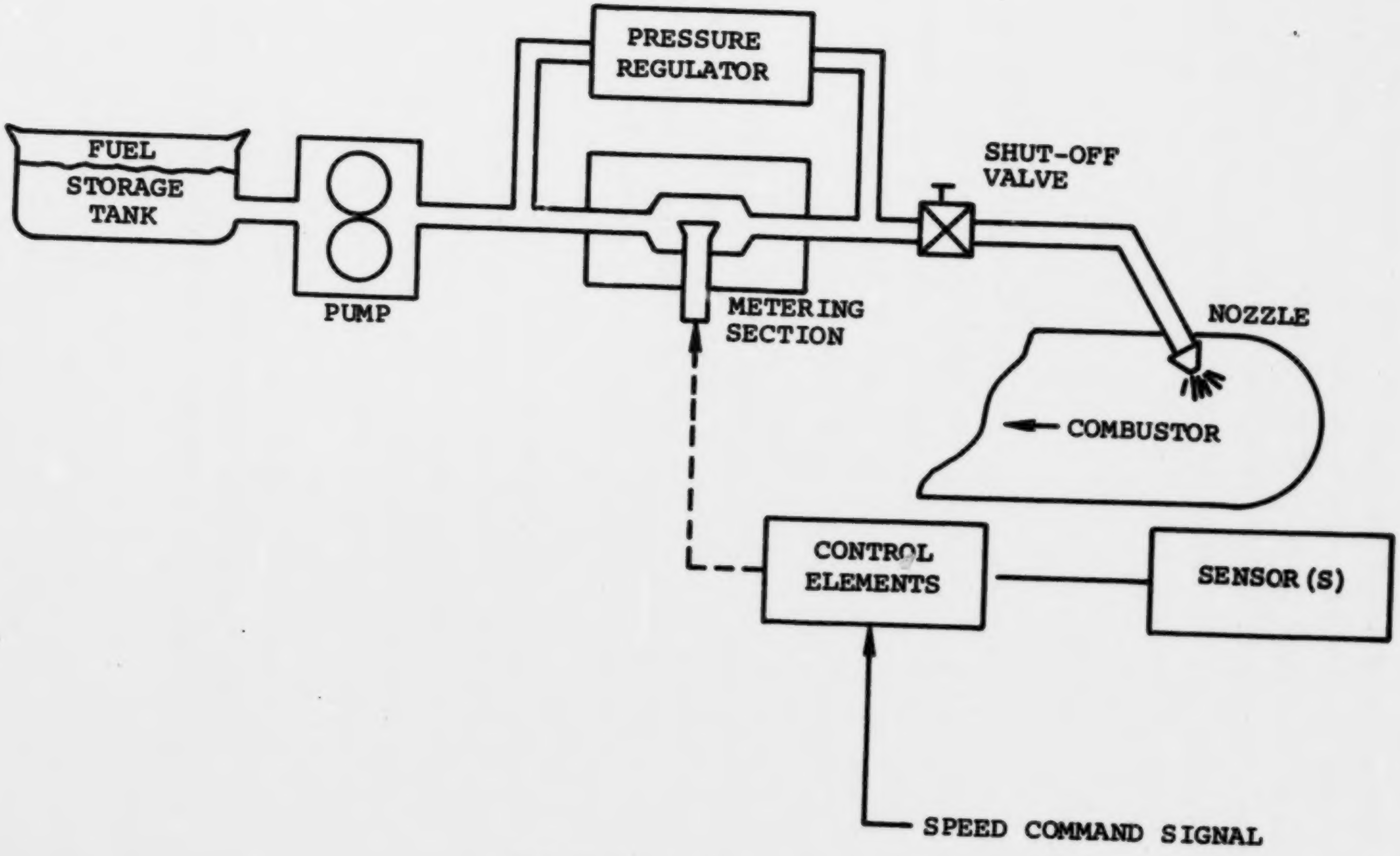
1.1.2 Configurations

Basic fuel control systems (Figure 3) fall within one of four major categories--hydromechanical, fluidic, electronic, or hybrid. The fuel tank, shut-off valve, nozzle, and often the pump are identical in each of the four. Major differences exist in the sensors, metering sections, and controller elements.

Hydromechanical control systems generally employ a rotating shaft as the speed sensor and rotating flyweights as the controller elements to position a metering valve. The desired speed demand is obtained through a mechanical arm that resets the load on the governor spring.

The fluidic control system normally has a pneumatic speed sensor element and pneumatic or hydraulic actuators to position the metering valve. Controller computations are accomplished fluidically (control of a fluid stream direction by another fluid stream). Speed inputs are usually accomplished by a mechanical arm that varies orifice size.

Electronic control systems have an electromagnetic speed sensing device, electronic controller elements, and an electromechanical or electrohydraulic metering section. Speed command signal is usually electrical.


PRESSURE REGULATOR
FUEL STORAGE TANK
SHUT-OFF VALVE
PUMP
METERING SECTION
NOZZLE
COMBUSTOR
CONTROL ELEMENTS
SENSOR (S)
SPEED COMMAND SIGNAL

BASIC FUEL CONTROL SYSTEM

FIGURE 3

Hybrid control systems may consist of various combinations of these three, thereby utilizing the most desirable design features of each.

Table 1 lists five trade-off parameters and their respective weighting factors. Weighting factors were chosen to indicate that initial cost, reliability, and maintainability (in that order) were more important as long as reasonable values for weight and size were obtained.

TABLE 1

CONTROL CONFIGURATION TRADE-OFF PARAMETERS
AND WEIGHTING FACTORS

Trade-off Parameters	Weighting Factor
Size	0.05
Weight	0.10
Initial Cost	0.40
Reliability (Life)	0.30
Maintainability	0.15

Initial cost was rated most important because of the necessity of keeping total vehicle initial cost within normal consumer budgets. Reliability was also important for relatively trouble-free operation and low-cost maintenance. The maintainability factor was reduced considerably with reliable components. To facilitate maintenance and reduce cost, a modular concept for fuel system components was recommended.

Economically, there was an optimum reliability for each item of equipment intended for a specific use. This was from the classical interrelationships between initial cost, maintenance cost, and reliability as shown in Figure 4. The actual optimum reliability figure for the selected system was determined as early as possible in the conceptual phase of the program trade-off studies. Weight and size were of less importance in this application, and designing for minimum weight and size might have unnecessarily raised component initial cost.

Table 2 lists the four major control configuration categories and compares relative size, weight, cost, reliability, and maintainability. The hydromechanical control configuration was used as a base, and the other configurations compared to it. In the trade-off parameters, a number greater than one signified size, weight, cost, reliability, and maintainability greater than that of the hydromechanical system, while numbers less than one indicated the opposite.

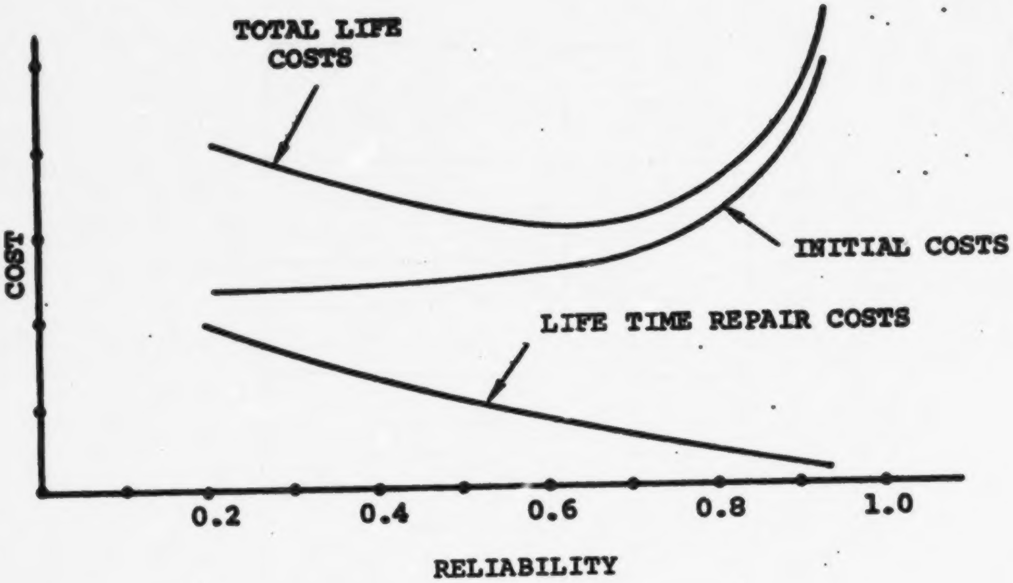
1.1.2.1 Fluidics

Referring to the 1971 technology, a comparison of fluidics with the base system indicated greater size, weight, and cost. Although composed of elements with no moving parts and tending toward high-density packaging, fluidics were rated greater than the base system because of the necessity for an external starting power supply and stringent filtration requirements. Since a fluidic control for this application was likely to use pneumatic power for control computation, several possibilities existed:

(a) An electric motor-driven air pump

(b) An engine-driven air pump

For increased life, either system could have been switched off once the engine reached idle speed, and pneumatic power could be derived from the engine compressor. However, compressor air was contaminated with dirt, exhaust emissions, free-carbon from the engine combustor,



OPTIMUM COST INTERRELATIONSHIP

FIGURE 4

TABLE 2

CONTROL CONFIGURATION TRADE-OFF COMPARISON

Parameters	Size		Weight		Cost		Reliability (Life)		Maintainability	
	1971	1975	1971	1975	1971	1975	1971	1975	1971	1975
Hydromechanical	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
Fluidic	1.1	1.0	1.0	1.0	1.2	1.0	0.7	0.9	0.7	0.9
Electronic	0.9	0.75	0.85	0.7	1.2	1.1	0.8	0.9	0.7	0.9
Hybrid	1.0	0.9	0.8	0.7	1.2	1.0	0.9	0.95	0.8	0.95

moisture, and small amounts of oil. Therefore, to protect the system, filtration would have to be available to remove a large portion of the contaminant.

Fluidic control devices for this application were considered less reliable and required more maintenance than the base system because of the power supply and contamination problems.

1.1.2.2 Electronics

A comparison of electronics with the base system indicates an electronic system will be smaller in size and weight but have a slightly increased cost. Size and weight savings were due to the utilization of solid-state devices, which accommodated high-density packaging. Although electronics required an external source of power, no additional component was needed as in fluidics, since a battery that was already available for starting the engine would be used as the power source.

Cost was normally greater than the base system in that the electronic control required a separate computing and power conditioning section, electronic/mechanical interface element, and signal transducers. However, cost was a relative quantity in this area and depended to a great extent on the complexity of the system to be controlled. For complex control systems, electronic utility was high because of the inherent ease of computation.

Electronics reliability was somewhat less than that of the base system because electronic components were more sensitive to the environment. This condition could be greatly alleviated by locating the computing section within the controlled environment of the automobile passenger compartment. In some applications, redundant electronic circuits were employed, providing greater reliability although costs increased.

Maintenance costs were higher when compared with the base system because of the additional components involved and specialized troubleshooting and repair equipment needed.

1.1.2.3 Hybrid

Trade-off parameters in hybrid control systems compared more favorably to the base system because the most desirable features of each configuration were combined. For instance, a fluidic valve could be utilized in the fuel system and eliminate the contamination problem previously cited for pneumatic through-flow systems. In this manner, the reliability of the fluidic element would be improved.

1.1.2.4 Summary

Referring to 1975 technology, general improvements in trade-off parameters are expected in fluidic and electronic devices, with fluidics showing a slight cost advantage over electronics. Hybrid control devices are also expected to improve with those in fluidics and electronics. Hydromechanical technology in 1975 is not expected to show significant advances over that of 1971.

1.2 FUEL CONTROL CHARACTERISTICS

Regardless of the implementation scheme employed (electronic, hydromechanical, etc.), basic fuel control generally consists of:

(a) A pump that routes filtered fuel to a metering section where a precise amount is automatically supplied to the engine according to the desired output power, while excess fuel flow is throttled back to pump inlet through the bypass valve. A key point is that the fuel must be automatically supplied to the engine to prevent compressor surge, turbine overtemperature, and overspeed. These items were considered too critical to leave to the judgment of the operator..

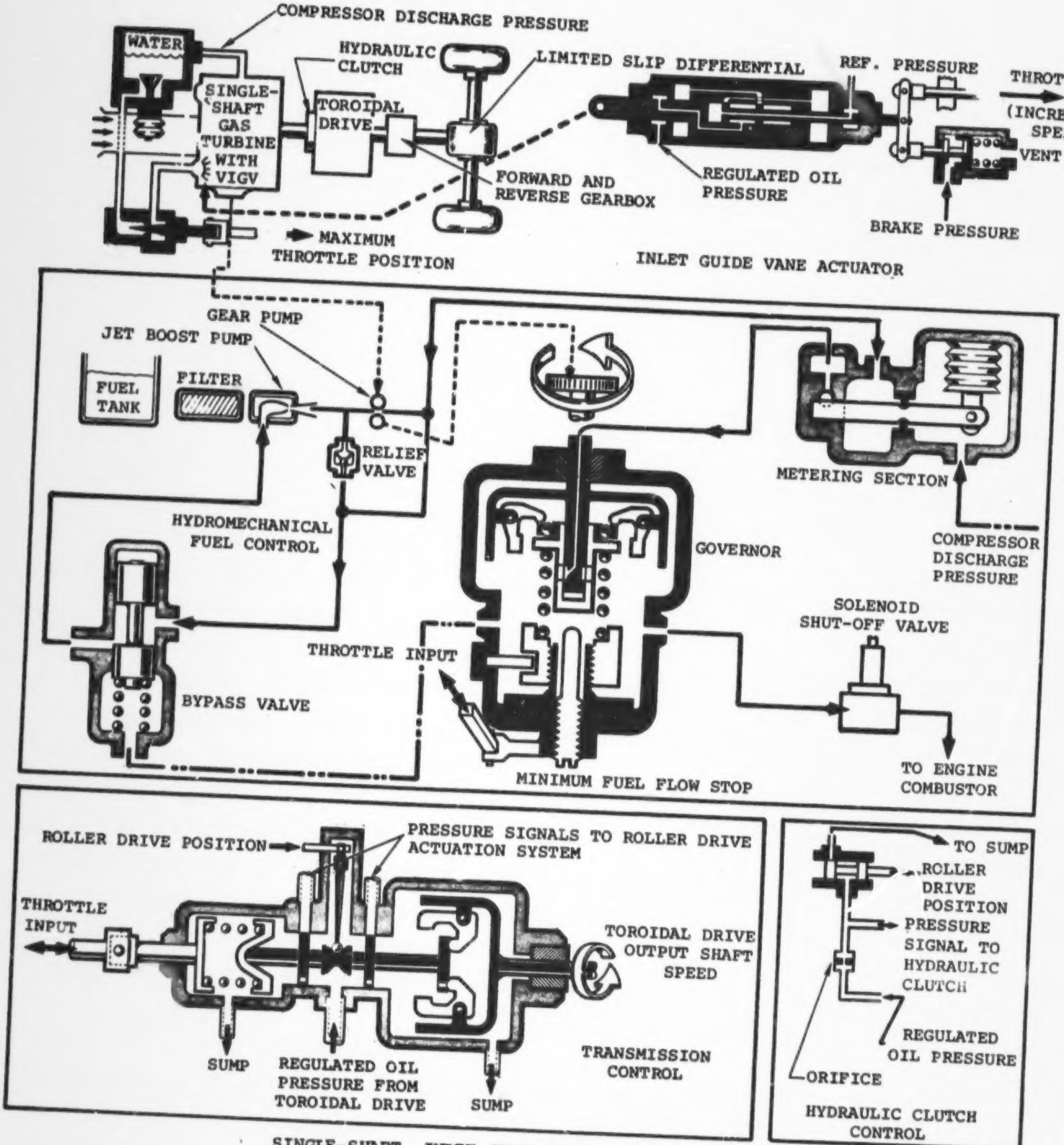
(b) A throttle reset governor, which either repositions or overrides the metering section, to control a desired engine speed.

(c) A fuel shut-off valve automatically controlled during start, with manual override for shutdown.

1.2.1 Hydromechanical

A cross-section schematic of a hydromechanical fuel control, suitable for controlling a single-shaft gas turbine is shown in Figure 5. The control has two orifices in series, with a single pump bypass valve controlling the pressure differential across both orifices. The upstream orifice is scheduled with compressor discharge pressure through an aneroid bellows, while the downstream orifice area is varied by a flyball governor sensing engine speed and throttle position. The throttle reset governor is equipped with a minimum fuel-flow stop to prevent engine burner blow-out during rapid deceleration. The fuel which passes through the bypass valve, in excess of the pump capacity, is used in a jet pump to boost the inlet pressure of the high-pressure pump, thereby preventing cavitation. The shut-off valve is a direct-acting, normally closed solenoid that receives signals from the automatic starting and protection control. A pressure relief valve is provided to prevent control overpressure, when the fuel system is dead-headed during shutdown.

The necessity for fuel schedule bias as a function of recuperator-out temperature, during engine start-to-idle operation, was considered in the study of the hydromechanical fuel control. Should this type of


COMPRESSOR DISCHARGE PRESSURE
WATER
SINGLE-SHAFT GAS TURBINE WITH VIGV
HYDRAULIC CLUTCH
TOROIDAL DRIVE
LIMITED SLIP DIFFERENTIAL
FORWARD AND REVERSE GEARBOX
REF. PRESSURE
THROTTLE (INCREASE SPEED)
VENT
REGULATED OIL PRESSURE
BRAKE PRESSURE
MAXIMUM THROTTLE POSITION
INLET GUIDE VANE ACTUATOR
GEAR PUMP
JET BOOST PUMP
FUEL TANK
FILTER
RELIEF VALVE
HYDROMECHANICAL FUEL CONTROL
METERING SECTION
COMPRESSOR DISCHARGE PRESSURE
GOVERNOR
SOLENOID SHUT-OFF VALVE
THROTTLE INPUT
BYPASS VALVE
MINIMUM FUEL FLOW STOP
TO ENGINE COMBUSTOR
ROLLER DRIVE POSITION
PRESSURE SIGNALS TO ROLLER DRIVE ACTUATION SYSTEM
THROTTLE INPUT
TOROIDAL DRIVE OUTPUT SHAFT SPEED
SUMP
REGULATED OIL PRESSURE FROM TOROIDAL DRIVE
SUMP
TRANSMISSION CONTROL
TO SUMP
ROLLER DRIVE POSITION
PRESSURE SIGNAL TO HYDRAULIC CLUTCH
REGULATED OIL PRESSURE
ORIFICE
HYDRAULIC CLUTCH CONTROL

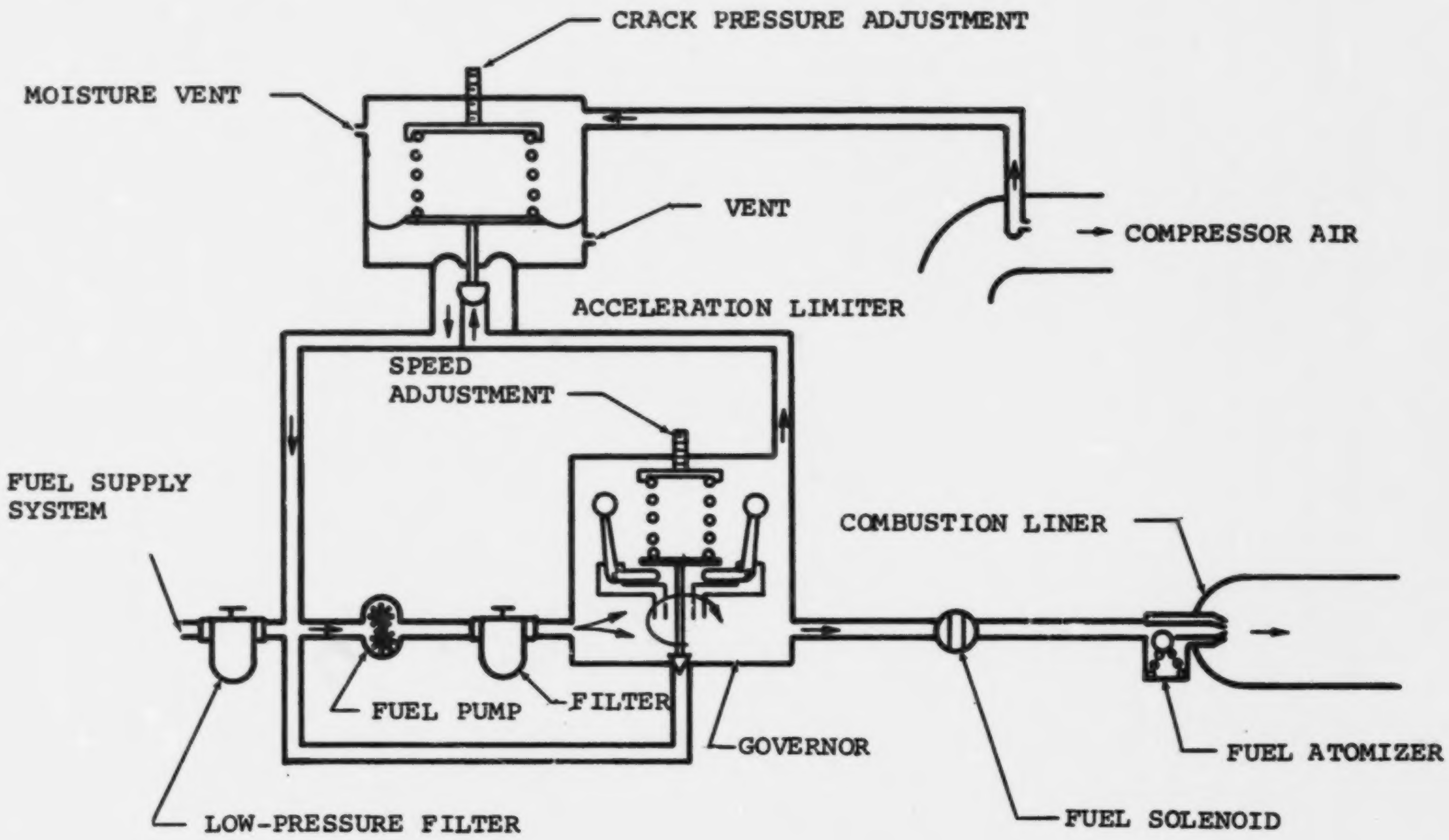
SINGLE-SHAFT, INLET GUIDE VANE, TOROIDAL DRIVE
FIGURE 5

control be necessary, the displacement output of a mechanical bimetallic temperature sensor could be used directly to provide additional control of start fuel-flow requirements. Further control studies may allow elimination of the aneroid bellows metering section in favor of a limiter-type scheduling control (Figure 6). The limiter control also acts as a pressure relief valve during shutdown, thereby eliminating an additional element.

1.2.2 Electronic

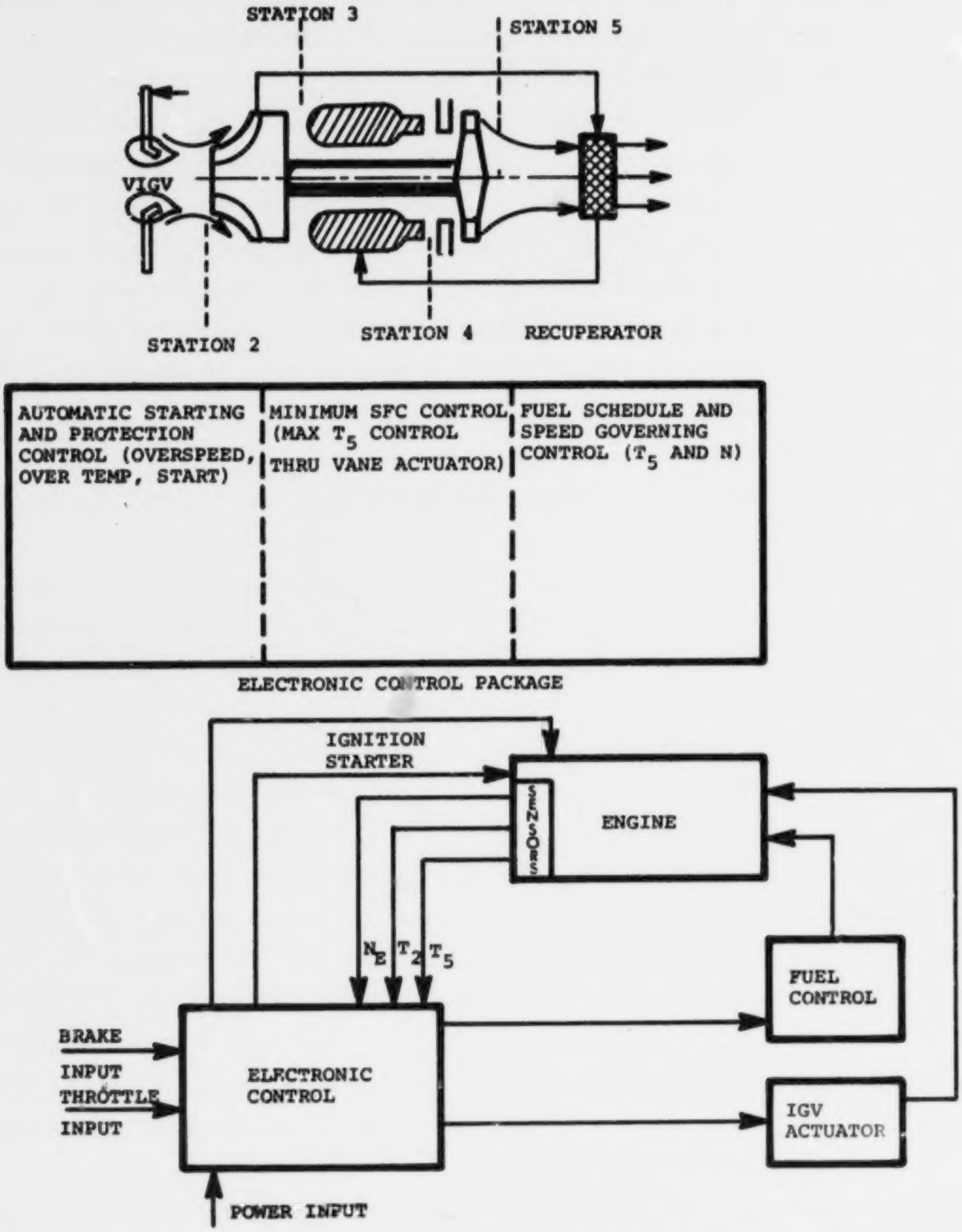
A functional schematic of an electronic control package is presented in Figure 7. This package consists of an automatic starting and protection control (common with the hydromechanical system), a minimum specific fuel consumption control, and a fuel schedule and speed-governing control.

The most important parameter to be controlled for a gas turbine engine is turbine inlet temperature, since this temperature is directly related to turbine material life, maximum power, maximum fuel economy, and compressor surge. Sensing and controlling turbine inlet temperature is readily accomplished by an electronic control, making it especially attractive for interfacing with a thermocouple temperature sensor. However, there are physical limitations to temperature sensor life set by temperature level, response time, material cost, and reliability. Because of these limitations, turbine exhaust temperature is generally selected for sensing due to the lower level and usually acceptable correlation with turbine inlet temperature. The sensing and control of turbine temperature is not restricted solely to electronics, as mechanical and fluidic sensors are available which could be utilized with hydromechanical or fluidic controls, respectively. However, one advantage offered by electronics (response-time compensation) is more easily implemented for the necessarily slower response and long-life temperature sensor.


CRACK PRESSURE ADJUSTMENT
MOISTURE VENT
VENT
COMPRESSOR AIR
ACCELERATION LIMITER
SPEED ADJUSTMENT
FUEL SUPPLY SYSTEM
COMBUSTION LINER
FUEL PUMP
FILTER
GOVERNOR
FUEL ATOMIZER
LOW-PRESSURE FILTER
FUEL SOLENOID

ACCELERATION LIMITER CONTROL

FIGURE 6


STATION 3
STATION 5
VIGV
STATION 2
STATION 4
RECUPERATOR
AUTOMATIC STARTING AND PROTECTION CONTROL (OVERSPEED, OVER TEMP, START)
MINIMUM SFC CONTROL (MAX T_5 CONTROL THRU VANE ACTUATOR)
FUEL SCHEDULE AND SPEED GOVERNING CONTROL (T_5 AND N)
ELECTRONIC CONTROL PACKAGE
IGNITION
STARTER
SENSORS
ENGINE
N_E
T_2
T_5
FUEL CONTROL
BRAKE INPUT
THROTTLE INPUT
ELECTRONIC CONTROL
IGV ACTUATOR
POWER INPUT

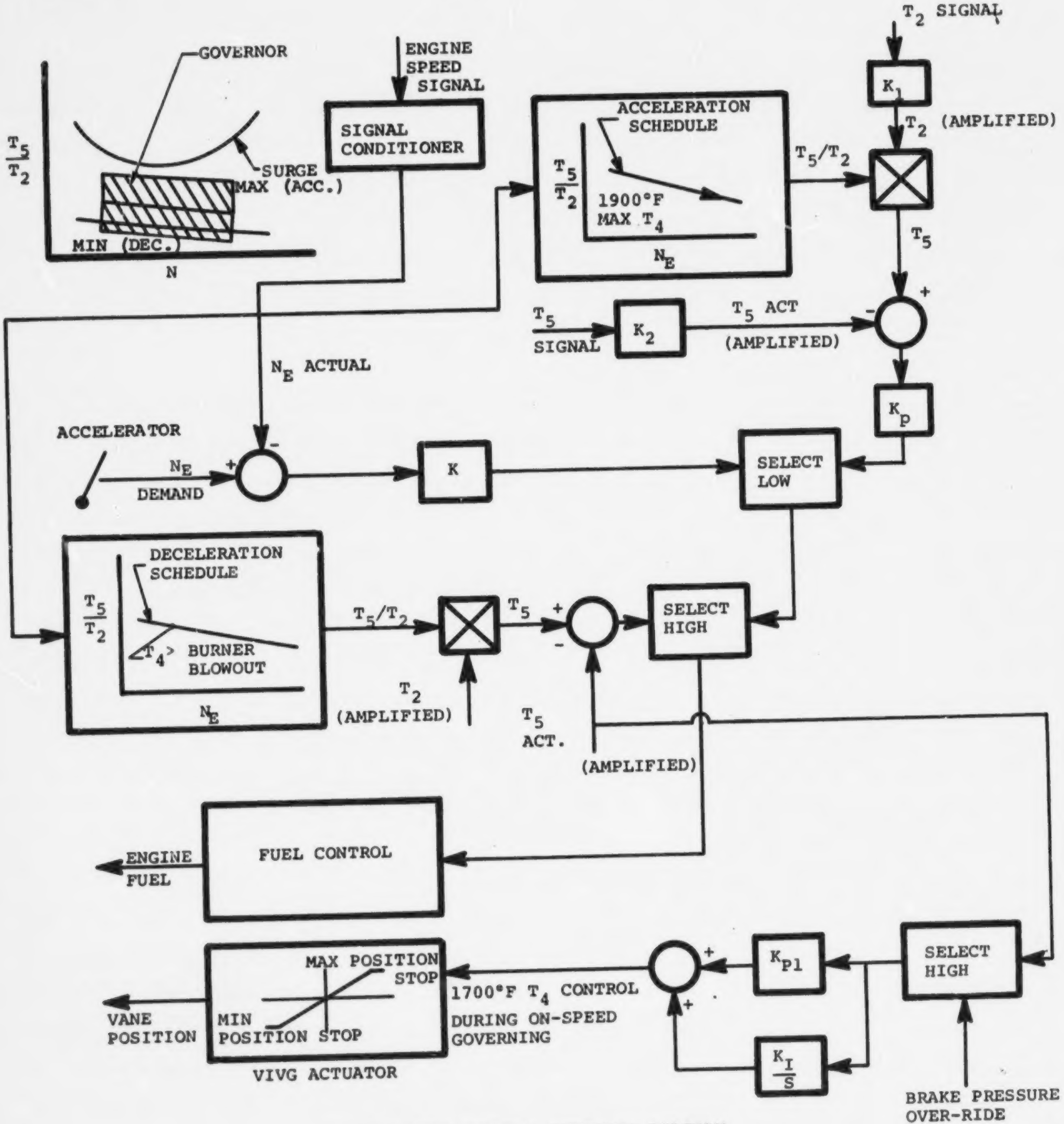
ELECTRONIC CONTROL SCHEMATIC FOR THE AUTOMOTIVE GAS TURBINE SINGLE-SHAFT VIGV

FIGURE 7

Referring to the fuel schedule and speed-governing portion of Figure 8, the fuel schedule control is closed-loop on turbine exhaust temperature (T_5) during acceleration and deceleration. The acceleration schedule is set at the maximum allowable T_5 consistent with maximum power requirements, while the deceleration schedule is set at a T_5 level to prevent burner blow-out. This low T_5 level during deceleration will aid in decreasing fuel consumption and deceleration time. Schedule bias with compressor inlet temperature (T_5) is only necessary for the acceleration schedule to prevent compressor surge during cold-day operation. The necessity of schedule bias for deceleration is not mandatory but is used because of its availability. However, this bias effect during deceleration is desirable, in that deceleration time over a wide range of T_2 variations is minimized.

For on-speed governing, a voltage proportional to speed is generated by a conditioned speed signal derived from an engine speed sensor (probably a monopole). During rapid acceleration when the speed demand is significantly higher than the actual speed, voltage is controlled to the fuel control driver by the acceleration schedule. As engine speed approaches demand speed, the speed error becomes less and, through a select low switch, blocks the acceleration schedule signal. At this point, engine speed will be maintained constant within some small error band unless the speed demand signal or engine load changes. During rapid deceleration, when speed demand is significantly lower than actual speed, the speed error is low and, through a select high switch, output is blocked in favor of the deceleration schedule signal. The deceleration signal remains in command until actual speed approaches demand speed, and normal speed governing is resumed.

An alternate and somewhat simpler approach for deceleration control is when a single, minimum value of fuel flow is set at 100-percent engine speed to prevent burner blow-out and the engine is allowed to decelerate on this amount of fuel. This scheme is consistent with


GOVERNOR
T_5/T_2
SURGE
MAX (ACC.)
MIN (DEC.)
N
ENGINE SPEED SIGNAL
SIGNAL CONDITIONER
ACCELERATION SCHEDULE
T_5/T_2
1900°F MAX T_4
N_E
T_2 SIGNAL
K_1
T_2 (AMPLIFIED)
T_5/T_2
T_5
+
−
T_5 SIGNAL
K_2
T_5 ACT (AMPLIFIED)
K_p
N_E ACTUAL
ACCELERATOR
N_E DEMAND
+
−
K
SELECT LOW
DECELERATION SCHEDULE
T_5/T_2
T_4 > BURNER BLOWOUT
N_E
T_5/T_2
T_5
+
−
SELECT HIGH
T_2 (AMPLIFIED)
T_5 ACT.
(AMPLIFIED)
ENGINE FUEL
FUEL CONTROL
VANE POSITION
MAX POSITION STOP
MIN POSITION STOP
VIVG ACTUATOR
1700°F T_4 CONTROL DURING ON-SPEED GOVERNING
+
+
K_{P1}
K_I/S
SELECT HIGH
BRAKE PRESSURE OVER-RIDE

ELECTRONIC CONTROL FUNCTION DIAGRAM
(MINIMUM SFC AND FUEL SCHEDULE CONTROL)
SINGLE-SHAFT VIGV ENGINE

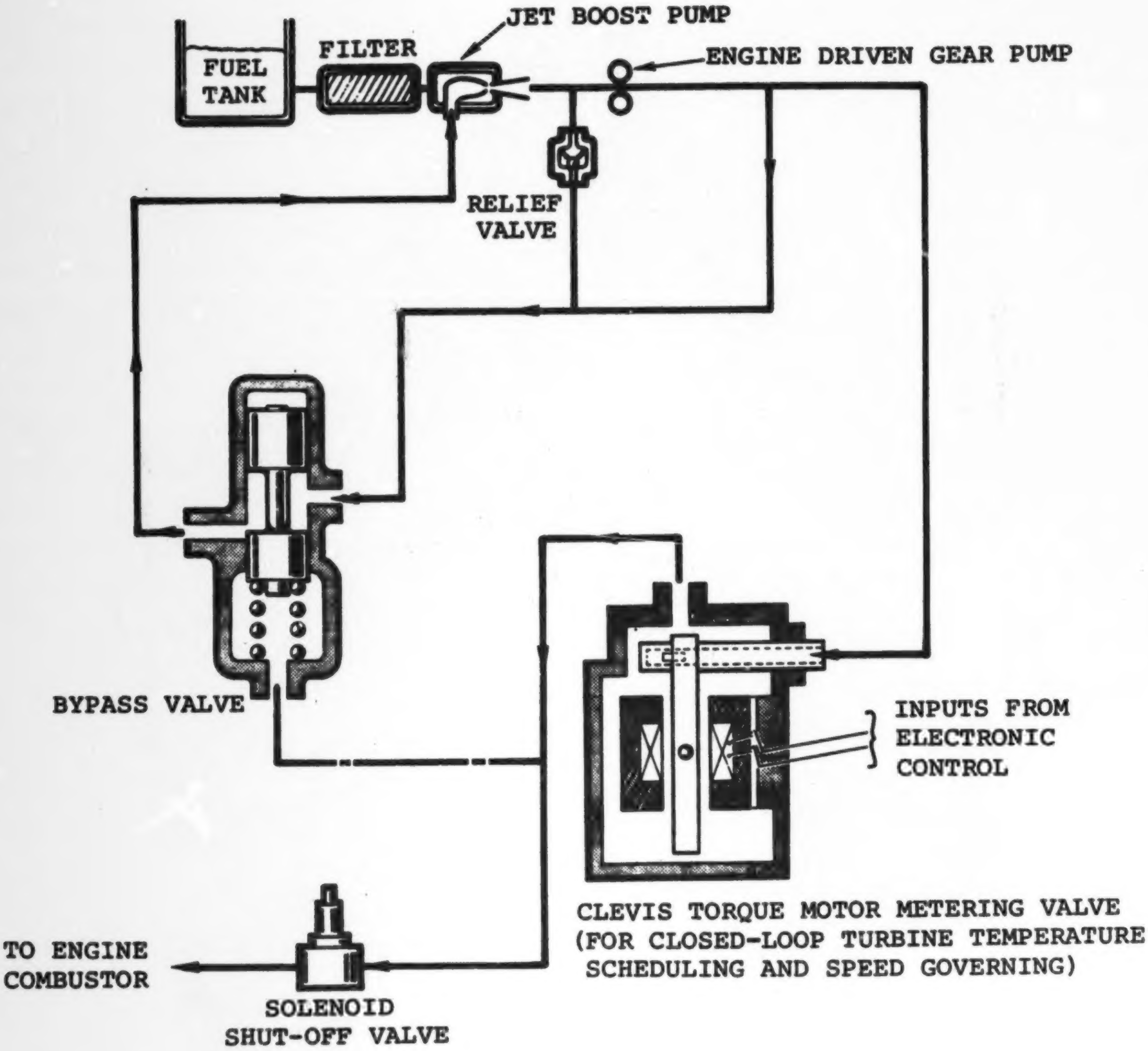
FIGURE 8

the hydromechanical method of deceleration control and is feasible because of the fairly narrow speed range between idle and full-throttle operation (60- to 100-percent speed). This is attractive because the deceleration control function generator is eliminated, resulting in reduced cost and increased reliability. The merits of the two deceleration schemes require further consideration of the recuperator effects. Final selection will be based on a trade-off study involving performance and cost-of-ownership.

A possibility for converting electronic fuel control signals into engine fuel flow is shown on Figure 9. This schematic resembles the hydromechanical fuel control, in which a filter, jet boost pump, high-pressure pump, bypass valve, relief valve, and solenoid shut-off valve are used. The bellows-operated metering section and throttle reset governor are replaced by a single electrohydraulic device for the conversion of electrical signals into engine fuel flow. This device (Figure 10) is a low-power (2 w max), direct-acting, balanced-clevis, shear valve operated by, and integral with, a torque motor armature. It is a fully developed unit of modular construction and may require only slight modifications to the metering port and electrical connector to become suitable for an automobile gas turbine system.

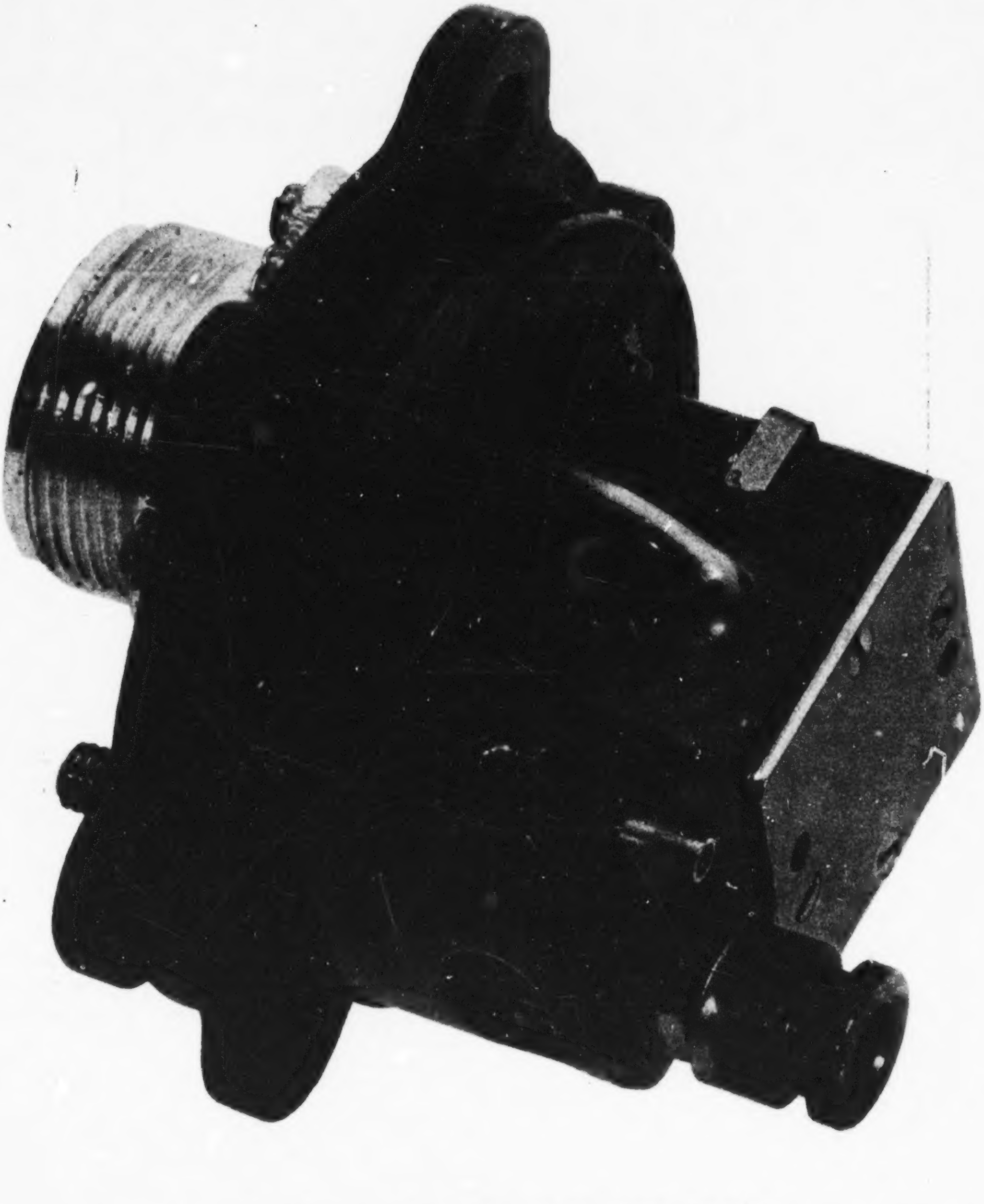
1.2.3 Fluidic

For the years 1972 to 1975, fluidic controls were not selected for application because of (1) the necessity for an external, probably pneumatic, starting power supply, and (2) their susceptibility to contamination. However, advances are being made in filtration and, by design, reduction of the contamination sensitivity can be accomplished. When these advances are realized, a fluidic temperature sensor (Figure 11) will be attractive because of direct turbine temperature measurement and the rugged design features of the temperature-sensing element. This sensor could be used with any method of control computation, provided that the proper interfacing elements are available.


JET BOOST PUMP
FUEL TANK
FILTER
ENGINE DRIVEN GEAR PUMP
RELIEF VALVE
BYPASS VALVE
INPUTS FROM ELECTRONIC CONTROL
CLEVIS TORQUE MOTOR METERING VALVE (FOR CLOSED-LOOP TURBINE TEMPERATURE SCHEDULING AND SPEED GOVERNING)
TO ENGINE COMBUSTOR
SOLENOID SHUT-OFF VALVE

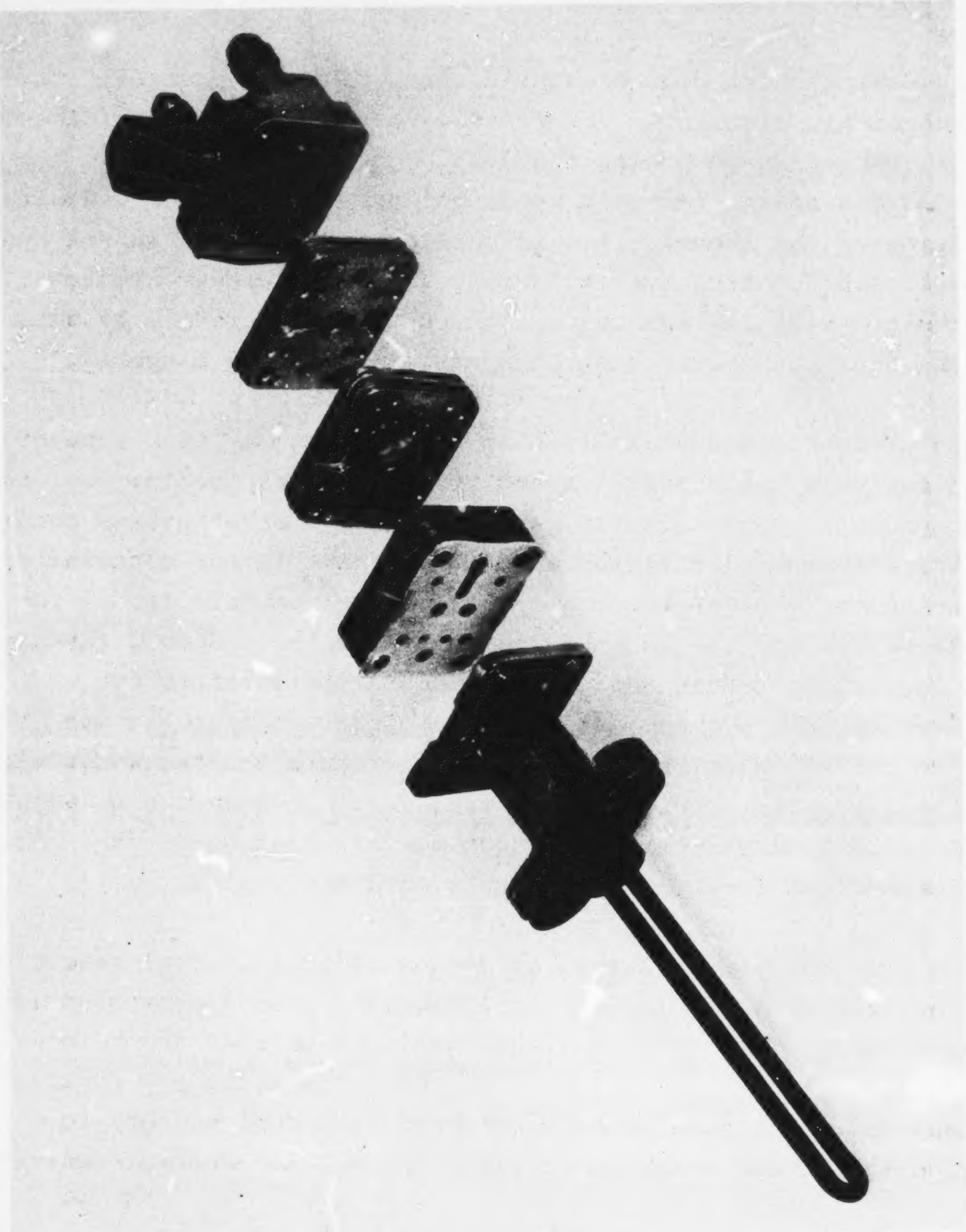
FUEL MANAGEMENT SCHEMATIC FOR ELECTRONIC FUEL CONTROL
(APPLICABLE TO SINGLE-SHAFT AND FREE-TURBINE ENGINES)

FIGURE 9



FUEL SCHEDULING AND GOVERNING MODULE
(ELECTRONIC CONTROL)

FIGURE 10



FLUIDIC TEMPERATURE SENSOR AND AMPLIFIER

FIGURE 11

1.2.4 Hybrid

Discussions have been presented throughout the preceding text to hydromechanical, electronic, or fluidic controls, referring directly to their respective computing sections. However, in reality, the total controls system for this application is composed of a desirable combination of the above. This is particularly evident in the hydromechanical fuel control system where solid-state electronics provide the automatic starting and protection control. Therefore by definition, the hydromechanical control system is actually a hybrid.

The trade-off study involved in determining the best type of fuel control for this application is not isolated solely to the fuel control components; consideration is also given to other system components. For instance, the single most important factor concerning hydromechanical-versus-electronic fuel control centers around the controlling device for the inlet guide vane actuator. Should the actuator be required to sense and control turbine temperature for part-throttle fuel economy, an electronic fuel control becomes attractive since the necessary high-quality temperature sensing would already be available. However, if part-throttle fuel economy can be adequately controlled by a direct-throttle input to the inlet guide vane actuator, a hydromechanical fuel control appears more attractive.

The previously established guideline of fuel control selection depending upon actuator implies that specific fuel consumption is an important factor in this system and should be weighed accordingly. However, factors such as initial cost and reliability are equally important and would have to be considered with fuel economy to determine the scheme which would yield the minimum cost-of-ownership.

In summary, hydromechanical and electronic controls offer the greatest potential for this system, with fluidic elements a possibility in discrete locations. Realistically, the decision between a hydromechanical and electronic control cannot be firmed until a detailed controls analysis is made, concerning those items discussed above.

1.3 INLET GUIDE VANE ACTUATOR

Variable compressor inlet guide vanes are used to minimize part-load fuel consumption. This is accomplished by setting the compressor inlet area at a value in which the turbine inlet temperature remains constant.

Figure 5 shows a throttle-operated, hydraulic-powered, follower-servo piston similar to existing power steering units which, through an external rod, moves the inlet guide vanes to a more open position as the accelerator pedal is depressed. The actuator is equipped with internal maximum and minimum position stops to prevent over-travel and compressor-surge, respectively. During deceleration, additional vehicle braking is provided by the engine through the vane actuator, i.e., when pressure is applied to the brake pedal, the vanes move to a more open position. Since the vehicle drive wheels are providing the energy through the drive gear train to rotate the engine during this mode, a more open vane position allows the engine compressor to induce additional air. The result will be greater braking torque imposed on the vehicle. This action is provided by a brake-actuated piston attached to a differential link so that the movement causes a repositioning of the vane actuator piston.

The vane-actuator control concept described above is somewhat simpler than the concept shown in Figure 8 but is an indirect means of controlling turbine temperature. Figure 8 features an electronically positioned actuator that senses and controls turbine temperature during part-throttle operation. During acceleration or deceleration, turbine temperature is set sufficiently high or low to prevent conflict with the part-throttle temperature control. For example, during a maximum power demand, a turbine inlet temperature (T_4) of 1900°F is reached that is 200°F above the part-throttle T_4. At this time, the actuator will move to a maximum-open position and remain there until the desired steady-state operating point is reached. Part-throttle operation is again resumed with the actuator modulating the vanes to maintain control of turbine temperature. A like control mode takes place during deceleration, except that the actuator moves the vanes to a minimum-closed position until part-throttle operation is resumed. Braking control is provided in a manner similar to that described for the direct-throttle input actuator, with the intelligence coming from a brake override signal through a select high switch located in the electronic actuator control circuitry.

The electronically controlled actuator would be similar to the drawing shown in Figure 5, except that movement of the servo piston would be through a direct-acting proportional solenoid or perhaps a hydraulic servo torque motor. These items would replace the differential linkage and its mechnical inputs, as depicted on Figure 5. The temperature-sensing element would be located in the turbine exhaust rather than the turbine inlet, to increase the life of the sensor.

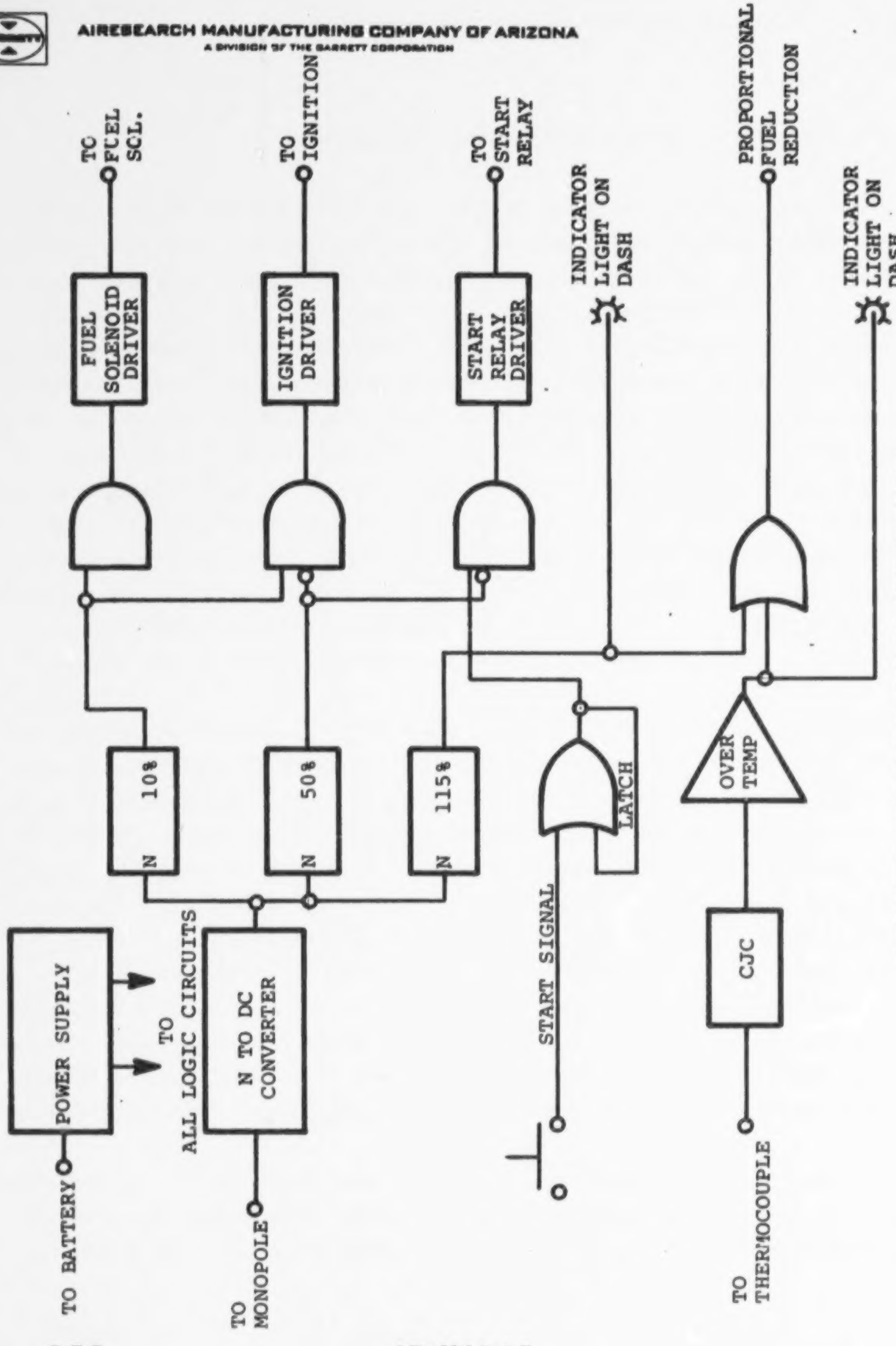
To define the actuator scheme best suited for this application, a detailed system analysis of various road and environmental conditions encountered during the life of the vehicle would be necessary. This would determine whether the cost and reliability of the simpler actuator would more than offset the savings in fuel consumption realized from the sensing and precise control of turbine temperature.

1.4 AUTOMATIC STARTING AND PROTECTION CONTROL

Unlike a spark-ignition engine, the gas turbine engine requires an automatic scheme for starting and engine protection. Gas turbine starting characteristics are greatly dependent upon combustor and fuel atomizer. For example, sensitivity to engine light-off is reduced considerably if fuel is supplied at a predetermined starter cranking speed. Engine protection is necessary because certain system malfunctions could result in overtemperature or overspeed. These two conditions are not significantly critical in a spark-ignition engine, because combustion takes place with the air-fuel ratio less than stoichiometric. Thus, an increase in fuel (failed carburetor) results in a decrease in combustor temperature. However, the opposite effect is true of the gas turbine engine. Also, overspeed (a run-away condition) is less likely in a spark-ignition engine, since the combustion chamber and valve train tend to limit the rotational speed of the engine.

Figure 12 is a logic diagram of a solid-state electronic automatic starting and protection circuit. Turbine temperature, speed, and a manual start-initiation switch are inputs, while output signals are provided to operate the starter relay, fuel solenoid, ignition unit, and overspeed and overtemperature indicator lights. A typical operating sequence is initiated by a momentary closure of the start switch, causing the start relay to pull in and latch. At 10-percent engine speed, the fuel solenoid and ignition unit are energized. At 50-percent engine speed, the starter relay and ignition unit are deenergized, thus dropping out the starter and ignition spark. Should an overspeed or overtemperature occur, the scheduled fuel will be reduced and an appropriate dash-mounted indicator light will illuminate.

Mechanical- and pneumatic-operated switching logic was considered for this application but discarded in favor of solid-state electronics. Mechanical switching is somewhat more complex as a speed drive and


TO BATTERY
POWER SUPPLY
TO ALL LOGIC CIRCUITS
TO MONOPOLE
N TO DC CONVERTER
N 10%
N 50%
N 115%
FUEL SOLENOID DRIVER
TO FUEL SCL.
IGNITION DRIVER
TO IGNITION
START RELAY DRIVER
TO START RELAY
INDICATOR LIGHT ON DASH
START SIGNAL
LATCH
TO THERMOCOUPLE
CJC
OVER TEMP
PROPORTIONAL FUEL REDUCTION
INDICATOR LIGHT ON DASH

AUTOMATIC START AND PROTECTION CIRCUIT

FIGURE 12

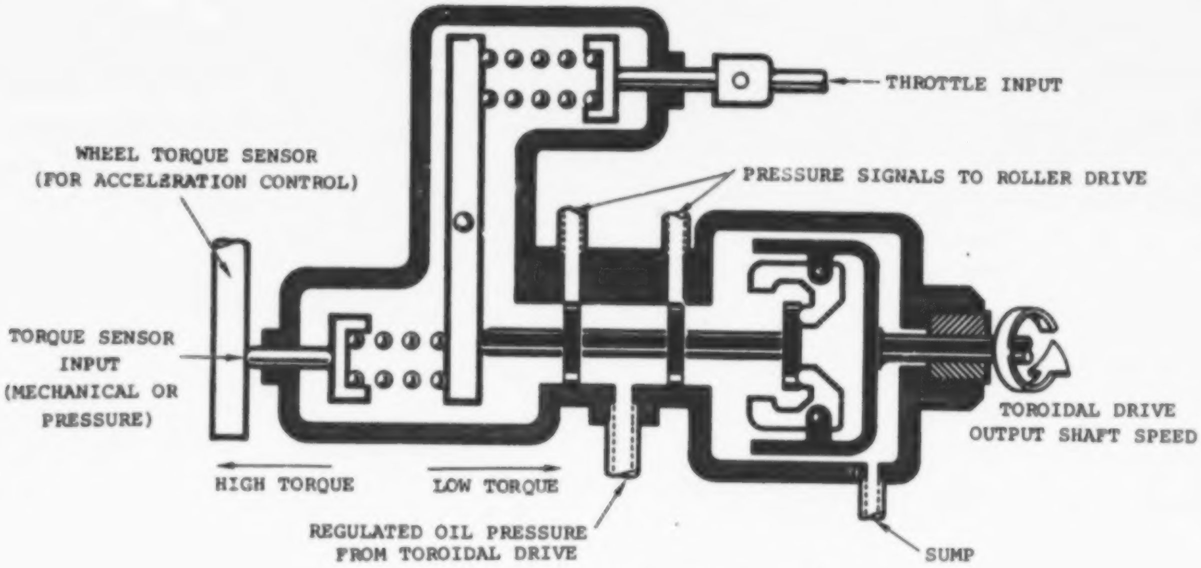
mechanical flyweights are necessary for sensing speed. For the pneumatic scheme, lack of available power necessitates an engine-rechargeable, air-storage bottle or a separate engine-driven air compressor. Also, both the mechanical or pneumatic schemes require micro switches to interface with the fuel solenoid. These are less reliable than solid-state switches under repeated cycling, a condition inherent in this system.

Functions of the automatic starting and protection control presented above constitute the minimum requirements for engine safety. The final modes of operation within the fault logic will be selected after a study of system reliability and driving safety hazards that might be present during a fault.

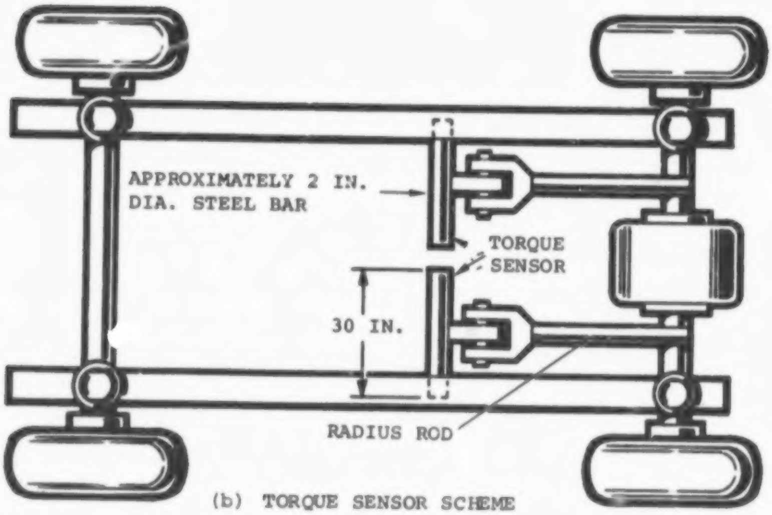
1.5 TRANSMISSION CONTROL

Several means of controlling the toroidal drive are currently being considered. The first scheme shown in Figure 5 controls the speed ratio within the toroidal drive as a function of throttle position and rear-wheel velocity. For example, to accelerate the vehicle, the throttle is depressed, resulting in greater fuel delivery to the engine, simultaneously causing the toroidal drive to downshift. As engine speed increases, the vehicle begins to accelerate. Higher vehicle speed is sensed by the speed servo that, in turn, begins to counteract the throttle input. This causes the toroidal drive to upshift to maintain the higher vehicle velocity at a lower engine speed. When cruising speed is reached, the toroidal drive will automatically be positioned in the lowest possible speed ratio, resulting in the lowest possible engine speed.

An alternate for controlling vehicle acceleration is Figure 13(a). Since drive-wheel torque is proportional to vehicle acceleration, the second scheme of Figure 5 features a drive-wheel torque sensor that


THROTTLE INPUT
WHEEL TORQUE SENSOR
(FOR ACCELERATION CONTROL)
PRESSURE SIGNALS TO ROLLER DRIVE
TORQUE SENSOR
INPUT
(MECHANICAL OR
PRESSURE)
TOROIDAL DRIVE
OUTPUT SHAFT SPEED
HIGH TORQUE
LOW TORQUE
REGULATED OIL PRESSURE
FROM TOROIDAL DRIVE
SUMP

(a) ACCELERATION AND CRUISE CONTROL


APPROXIMATELY 2 IN.
DIA. STEEL BAR
TORQUE
SENSOR
30 IN.
RADIUS ROD

(b) TORQUE SENSOR SCHEME

VEHICLE ACCELERATION CONTROL

FIGURE 13

automatically selects the optimum toroidal-drive speed ratio for maximum acceleration. Mechanically, this scheme functions identically as that of Figure 5, but the toroidal-drive, speed-ratio selection is biased by vehicle acceleration. (This may be thought of as a control system that is closed-loop on vehicle acceleration.)

One possible concept for the torque sensor is shown on Figure 13(b). This scheme consists of a torque rod, one end of which is attached to the frame of the auto and the opposite end to the drive wheels through the axle housing. As the vehicle is accelerated, the vehicle drive-wheel thrust load displaces the center of the torque rod, thus creating a displacement signal proportional to vehicle acceleration.

The merits of torque sensing to control vehicle acceleration were not thoroughly evaluated. Examination of this scheme would involve a computer study to optimize vehicle acceleration through the sensing and control of drive-wheel torque. However, should this study show significant gains, the torque-sensing scheme would be further explored.

Since the transmission is a traction drive and is mechanical in nature, a mechanical control for changing speed ratio is more feasible because of the ease of interfacing. Items such as permanent magnet generators and monopoles were considered for transmission speed sensors but ruled out in favor of the mechanical scheme.

1.6 HYDRAULIC CLUTCH CONTROL

The clutch control (Figure 5) consists of a hydraulic servo spool that senses the position of the transmission traction rollers and controls the muscle pressure to the clutch actuator when the vehicle approaches low-speed operation. This causes the clutch to slip, while transmitting some portion of torque to the rear wheels. This operation is necessary to allow the engine to idle at a low vehicle velocity.

Under consideration, but not shown, is the probable necessity for an additional input to the clutch control. This input might consist of a turbine speed input or an output-shaft-speed interlock timing valve that would allow the vehicle to have adequate "curb" pulling power. For example, suppose an automobile equipped with the type of clutch control shown in Figure 5 pulled square with a curb and then attempted to start from rest and pull over it. As the engine throttle was depressed, only a fixed amount of torque would be transmitted to the drive wheels since the vehicle has zero forward velocity. Depending on curb height, this fixed amount of drive-wheel torque might not be sufficient to move the vehicle forward. However, if there were feedback from engine speed, or an equivalent, to decrease clutch slippage and increase drive-wheel torque, the maximum engine power could be utilized to pull over the curb.

A mechanical device was selected in this application because of simplicity and ease of interfacing with the hydraulically operated clutch.

1.7 MAXIMUM POWER CONTROL

The purpose of the maximum power control (water injection) is to boost engine horsepower when maximum acceleration is desired. Referring to Figure 5, the maximum power control senses T_2 and maximum-throttle position to regulate the amount of water injected into the engine. Compressor discharge pressure, P_3, is used to pressurize the water tank, and water is injected through fixed-area nozzles to control the injection rate as a fixed percentage of engine airflow. A bimetal temperature sensor and valve combination unit is mounted in the compressor inlet duct and allows water to flow to the injector nozzles through a throttle-operated needle valve on hot-day conditions only.

A mechanical temperature sensor was selected because (1) it can be combined easily with a shut-off feature, and (2) high-speed response is not necessary. This decision may require re-examination based upon the final selection of an engine fuel control, since the electronic version also requires a T_2 sensor. The throttle-operated needle valve was selected because of its simplicity.

1.8 SENSORS

Sensors for the control systems previously discussed consist of devices that can measure turbine speed, N_E, turbine exhaust temperature, T_5, and compressor inlet temperature, T_2. The speed sensor will probably be a monopole which is a proximity, ac-type, single-pole generator with output frequency proportional to speed. Because this type of sensor does not require an external drive and is relatively simple in construction, long service-life can be expected. When used in systems where a dc voltage level is required, the monopole signal is conditioned by a solid-state rectifying and filtering circuit.

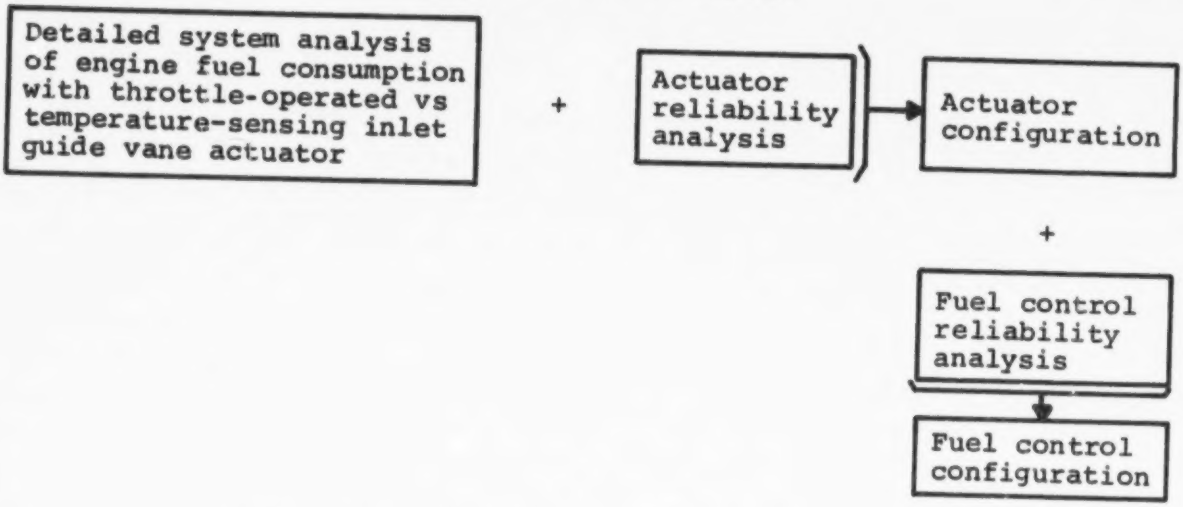
Two sensors are candidates for sensing T_2--a bimetal mechanical displacement device and an electrical resistance device, which produces a voltage change resistance in proportion to temperature. Because both devices are suitable for this application, the selection of one or both depends upon the fuel control and water injection control.

Sensing T_5 is more difficult than T_2 because of the elevated temperature. Three possibilities exist--a mechanical displacement quartz rod, a thermocouple, or a fluidic device. A quartz rod sensor looks attractive for a topping device (possibly for sensing overtemperature in the hydromechanical control system), while a thermocouple can be used for topping or temperature modulation. A thermocouple is especially suited for an electronic temperature control because of the ease of interfacing. The fluidic temperature sensor is attractive because of its rugged sensing element, and because it can be used with hydromechanical or electronic controls with proper interfacing devices.

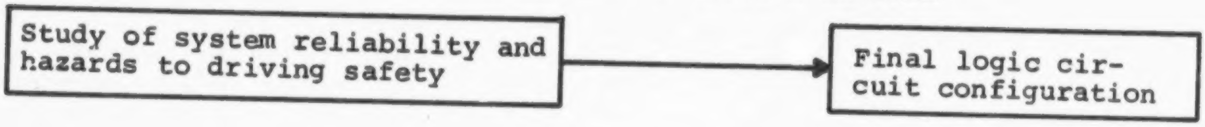
1.9 SUMMARY OF TASKS TO DETERMINE FINAL CONTROL SYSTEM CONFIGURATION

The following series of flow diagrams summarize the tasks necessary to determine the final system configuration for the single-shaft gas turbine:

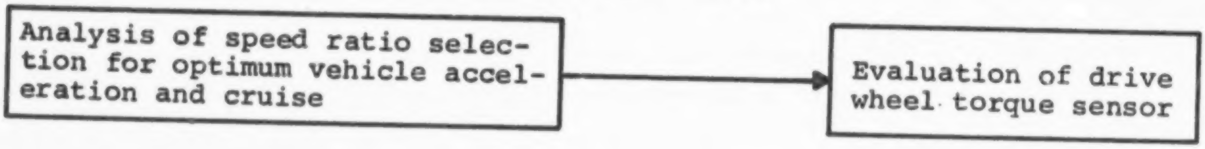
FUEL MANAGEMENT SYSTEM


Detailed system analysis of engine fuel consumption with throttle-operated vs temperature-sensing inlet guide vane actuator
+
Actuator reliability analysis
Actuator configuration
+
Fuel control reliability analysis
Fuel control configuration

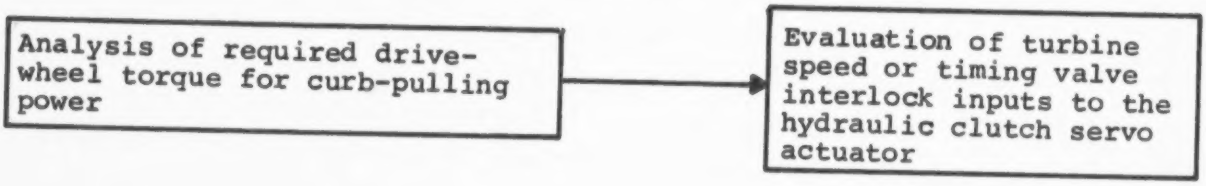
AUTOMATIC STARTING AND PROTECTION CONTROL


Study of system reliability and hazards to driving safety
Final logic circuit configuration

TRANSMISSION CONTROL


Analysis of speed ratio selection for optimum vehicle acceleration and cruise
Evaluation of drive wheel torque sensor

HYDRAULIC CLUTCH CONTROL


Analysis of required drive-wheel torque for curb-pulling power
Evaluation of turbine speed or timing valve interlock inputs to the hydraulic clutch servo actuator

1.10 FUEL CONTROL SYSTEM COMPONENT CONSTRUCTION

1.10.1 Hydromechanical

The metering section, governor, jet boost pump, and relief valve will be integrated in a single housing, whereas the bypass valve, solenoid valve, fuel pump, and inlet filter will be separate modules bolted to the main housing. This packaging arrangement was chosen to minimize raw-material cost and provide ease of maintainability.

With the exception of the bypass valve housing, all main housings will be die-cast aluminum with internally drilled fluid passages arranged in a manner that will eliminate compound-angle drilling. Experience has shown that the bypass valve and governor shaft must be constructed of a hard, corrosion-resistant material. This is necessary to minimize metering-edge erosion and surface oxidation (rust) caused by water entrained in the fuel. A 400-series stainless steel complies with these requirements and is relatively inexpensive because of its adaptability to casting.

The fuel pump gears will probably be constructed from sintered metal or molded plastic to minimize machining cost and provide built-in lubricating for increased wear resistance.

The metering valve bellows will be hydroformed from either commercial brass or phosphorous bronze. These materials are inexpensive and offer good corrosion resistance.

Miscellaneous hardware, such as springs and adjustment screws, will be made of 17-7 and 17-4 pH, respectively. This selection was based on the necessity for high strength, hardness, and corrosion-resistance.

1.10.2 Electronic Control Assembly

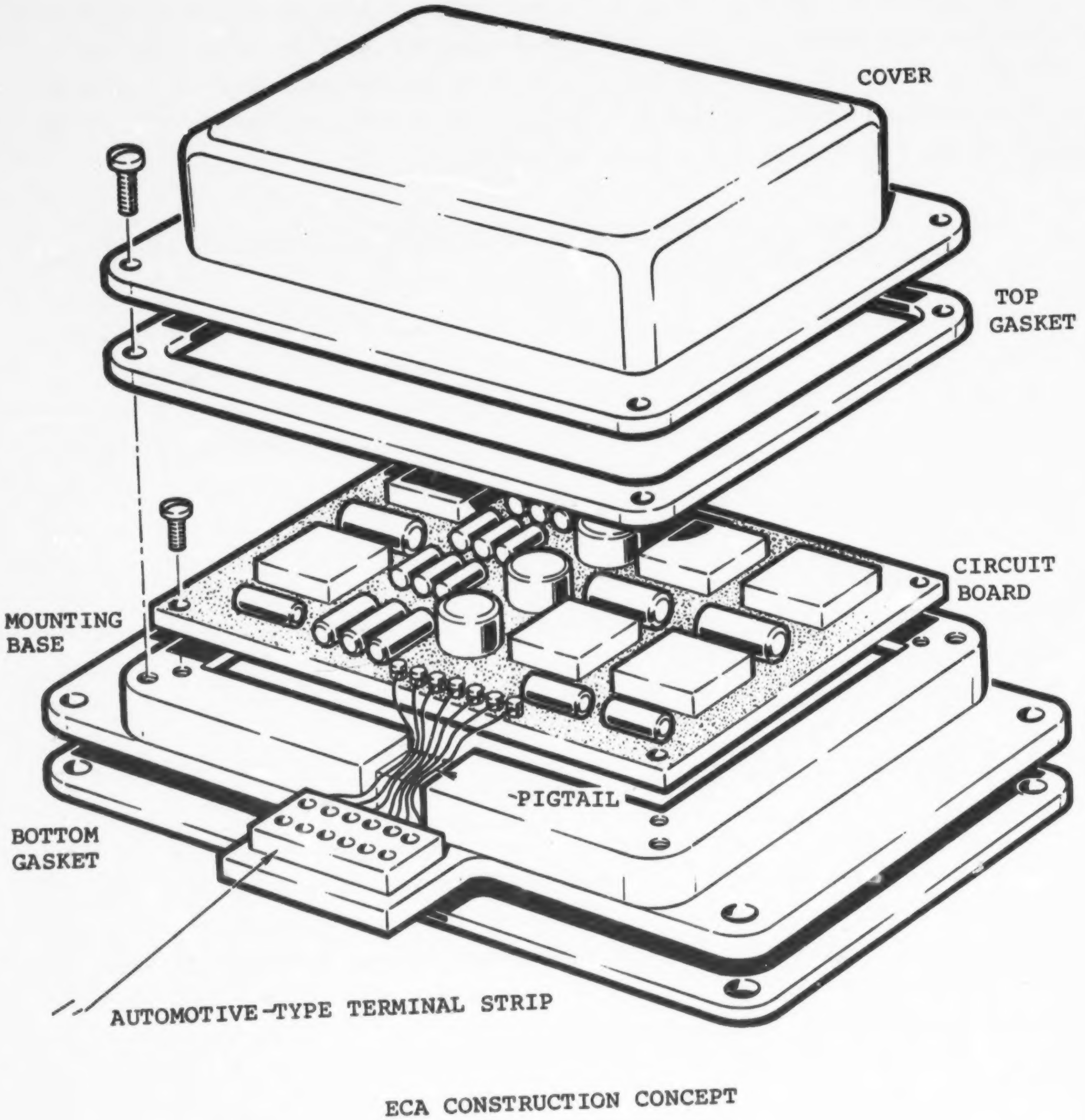
Studies indicate that construction of the electronic control assembly (ECA) should be along conventional lines, as typified by the system presented in Figure 14. This is consistent with present state of the art techniques in manufacturing printed circuit boards.

Typical construction (Figure 14) will probably consist of a mounting base of stamped sheet metal that bolts to a convenient flat surface within the compartment; a rubber bottom gasket seals the under part of the mounting base.

The electronic circuits are contained on a single printed circuit board. Electrical interface is accomplished by pigtail solder connections from the printed circuit board to an automotive-type terminal strip; the strip is exposed for interface connections even with the cover in place. The cover is also of stamped sheet metal construction and is retained by screws on the mounting base. A rubber top gasket provides a weather seal between the cover and base.

Circuit-board size will be approximately 6 x 4 in. and will include a mixture of discrete components (resistors, diodes, transistors, etc.) and integrated circuits. The circuit board will contain the automatic starting and protection, variable inlet guide vane control, fuel governor set-point control, and signal conditioner circuits. These circuits will operate from a 12-vdc battery power source input through suitable filter and regulator circuits also contained on the printed circuit. Electronic scheduling and gains will be accomplished by using medium-scale, linear, integrated circuits.

Speed logic decision circuitry will use medium scale integration (MSI) voltage comparators in conjunction with digital integrated circuit (IC) logic packs and high-powered output driver stages. All of


COVER
TOP GASKET
CIRCUIT BOARD
MOUNTING BASE
PIGTAIL
BOTTOM GASKET
AUTOMOTIVE-TYPE TERMINAL STRIP

ECA CONSTRUCTION CONCEPT

FIGURE 14

the MSI circuits are available, off-the-shelf items in dual-in-line packages. The present trend is toward higher packaging densities; quad op-amps (four separate operational amplifiers within a single package) are presently available in a single, dual-in-line package. Similar quad voltage comparators are also available. Prices forecast for these quad units indicate that they will sell for less than $1.00 per package in large quantities by 1973.

1.10.3 Inlet Guide Vane Actuator; Transmission Control; Hydraulic Clutch Control

These items will be operating in a closed-loop oil system; therefore, the use of corrosion-resistant materials is not necessary. Meehanite housings and 52100 steel valves are considered the best material combination, plus availability, low cost, and reliable operation.

1.10.4 Maximum Power Control

The throttle-operated needle valve and housing will be of molded plastic. The housing will be equipped with standpipe interfaces suitable for external, flexible-hose, band-clamp connections. This type of construction is best suited for high-volume, low-cost production.

1.10.5 Sensors

The typical rod and tube-type temperature sensor consists of a rod of low-expansion material (quartz) attached to a tube of high-expansion material (inconel). When used as a topping device, the differential output of the sensor normally closes a micro-switch. The micro-switch mounting is insulated from the temperature probe by minimizing the heat conduction path.

The thermocouple probe for sensing T_5 will be a shielded chromel-alumel junction. Because of the temperature variation of the electronic control, a cold-junction compensation will be incorporated. The thermocouple will be connected to a differential amplifier in the electronic control, thereby ensuring insensitivity to thermocouple resistance and leakage to ground.

The fluidic T_5 sensor is of the sonic oscillator-type, with the temperature probe constructed of a coiled, thin-walled tube submerged in the exhaust gas stream. The temperature signal is pneumatic and is transmitted to the signal-conditioning fluidic elements mounted on the external boss of the temperature probe. The conditioning section is made of thin, stainless-steel laminants brazed together to form a computing stack. The output signal is extracted from the computing stack through a threaded-type fluid connection.

The mechanical T_2 sensor is comprised of a stack of bimetal discs. Contraction and expansion of these discs is transmitted directly through a brass rod that acts as a valve to register the water supply for water injection.

The speed sensor will be a monopole wound with heavy-gauge wire, which ensures good reliability and good signal strength for a wide variation in pole air-gap.

2. CONTROL SYSTEM DESCRIPTION FOR THE FREE-TURBINE POWERED VEHICLE

The control system functional schematic for the free-turbine powered vehicle is presented in Figure 15. The primary purpose of this schematic is to present a total system concept similar to that discussed for the single-shaft engine. When compared with the single-shaft engine, controlling elements are essentially alike; the major difference being in the transmission control. This difference is a result of the three-speed automatic transmission and the more difficult problem of preventing free-turbine overspeed.

2.1 CONTROL SYSTEM COMPONENTS

Six major elements comprise the control system:

(a) Fuel control

(b) Turbine nozzle actuator

(c) Automatic starting and protection control

(d) Transmission control

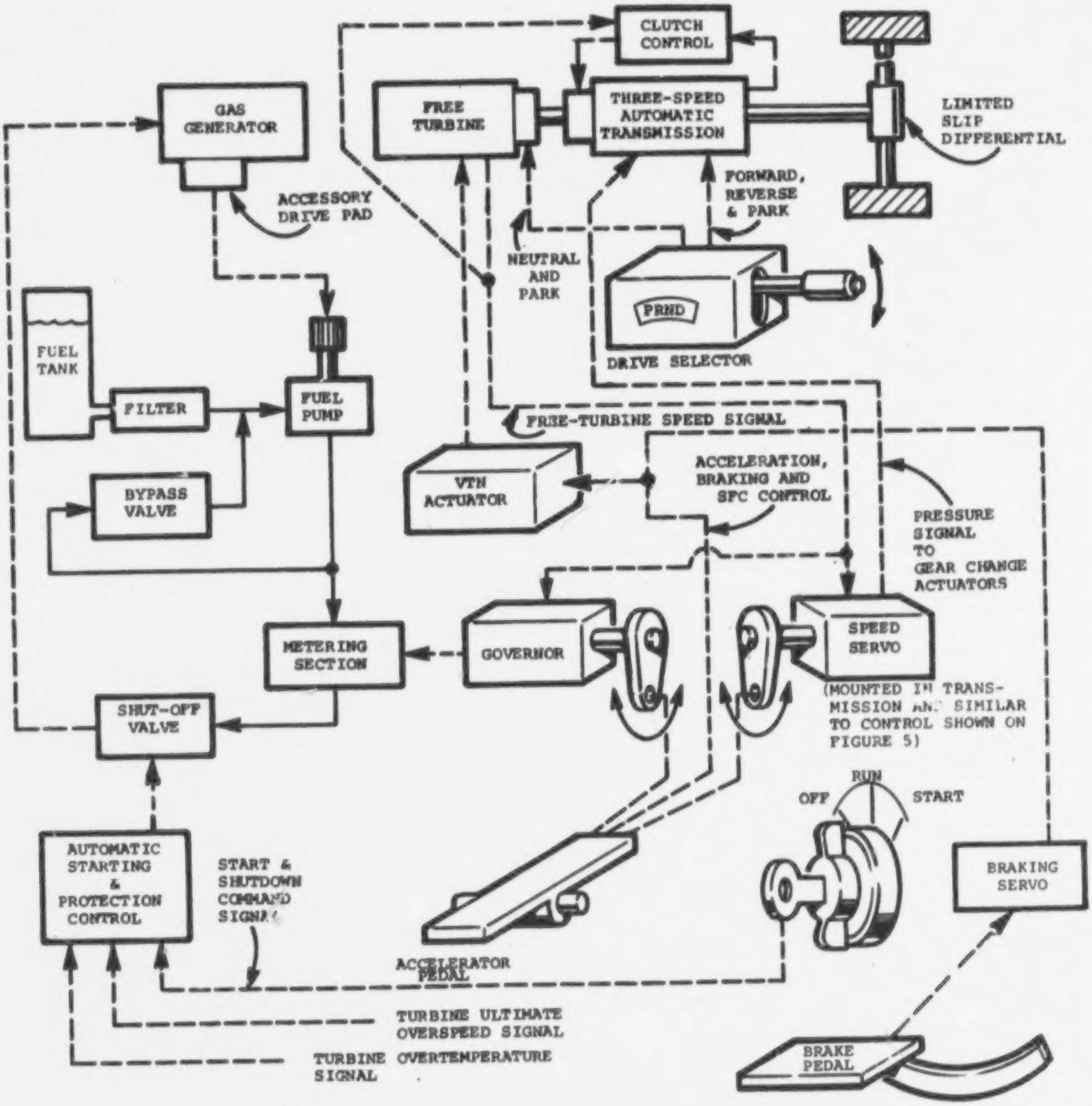
(e) Hydraulic clutch control

(f) Sensors

Several implementation schemes were studied for these elements and are presented in the text that follows. Figure 16 is a schematic representation of the control components.

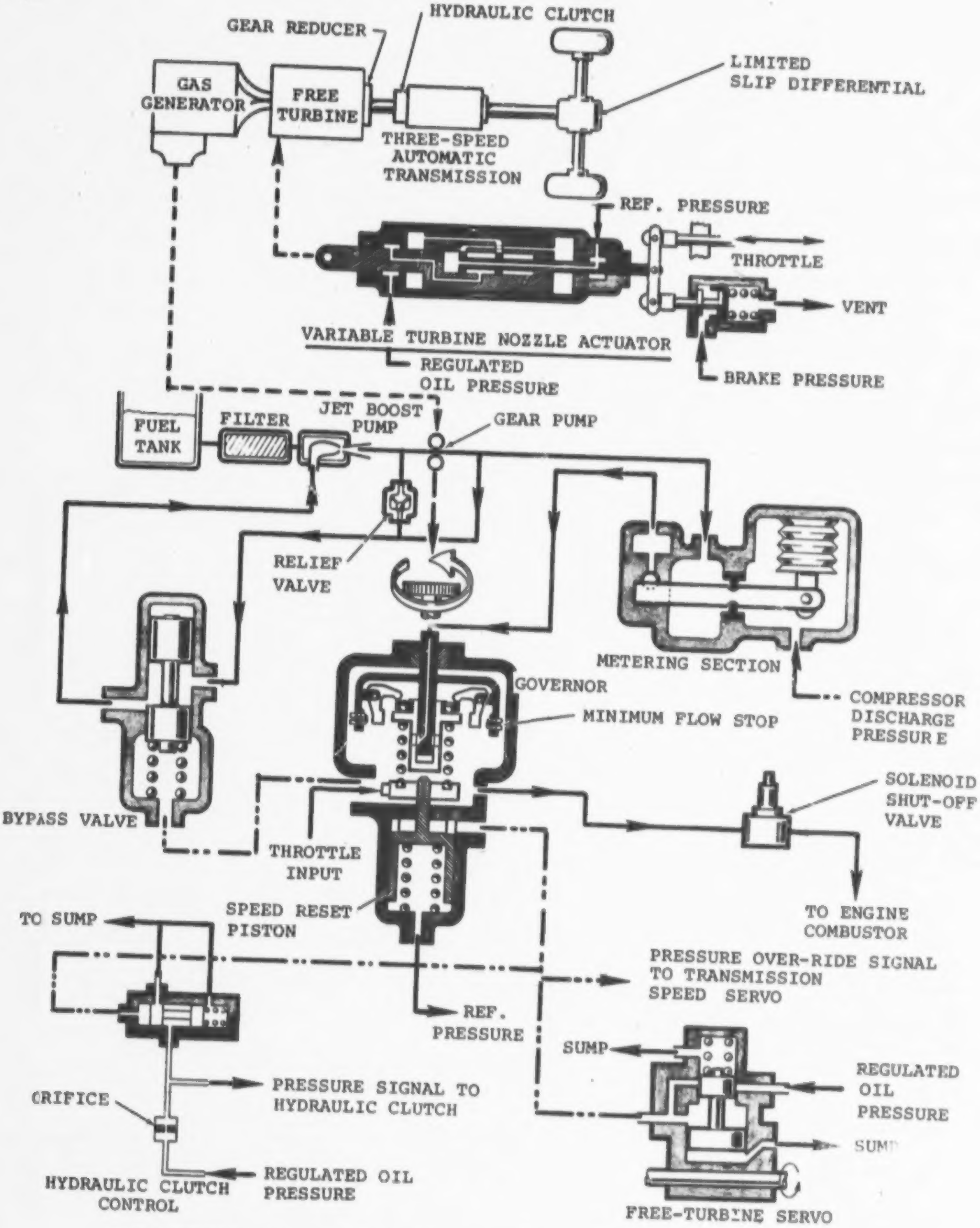
2.1.1 FUEL CONTROL

The basic fuel control is similar to that of the single-shaft turbine except that the throttle reset governor is biased by a signal that prevents overspeed in the event of loss of load. This scheme precludes free-turbine damage without shutting down the gas generator.


CLUTCH CONTROL
GAS GENERATOR
FREE TURBINE
THREE-SPEED AUTOMATIC TRANSMISSION
LIMITED SLIP DIFFERENTIAL
ACCESSORY DRIVE PAD
FORWARD, REVERSE & PARK
NEUTRAL AND PARK
PRND
DRIVE SELECTOR
FUEL TANK
FILTER
FUEL PUMP
BYPASS VALVE
FREE-TURBINE SPEED SIGNAL
VTN ACTUATOR
ACCELERATION, BRAKING AND SFC CONTROL
PRESSURE SIGNAL TO GEAR CHANGE ACTUATORS
METERING SECTION
GOVERNOR
SPEED SERVO
(MOUNTED IN TRANSMISSION AND SIMILAR TO CONTROL SHOWN ON FIGURE 5)
SHUT-OFF VALVE
OFF
RUN
START
AUTOMATIC STARTING & PROTECTION CONTROL
START & SHUTDOWN COMMAND SIGNAL
BRAKING SERVO
ACCELERATOR PEDAL
TURBINE ULTIMATE OVERSPEED SIGNAL
TURBINE OVERTEMPERATURE SIGNAL
BRAKE PEDAL

CONTROL SYSTEM FUNCTIONAL SCHEMATIC

FIGURE 15


GEAR REDUCER
HYDRAULIC CLUTCH
GAS GENERATOR
FREE TURBINE
LIMITED SLIP DIFFERENTIAL
THREE-SPEED AUTOMATIC TRANSMISSION
REF. PRESSURE
THROTTLE
VENT
VARIABLE TURBINE NOZZLE ACTUATOR
REGULATED OIL PRESSURE
BRAKE PRESSURE
FUEL TANK
FILTER
JET BOOST PUMP
GEAR PUMP
RELIEF VALVE
METERING SECTION
GOVERNOR
MINIMUM FLOW STOP
COMPRESSOR DISCHARGE PRESSURE
SOLENOID SHUT-OFF VALVE
BYPASS VALVE
THROTTLE INPUT
SPEED RESET PISTON
TO ENGINE COMBUSTOR
TO SUMP
PRESSURE OVER-RIDE SIGNAL TO TRANSMISSION SPEED SERVO
REF. PRESSURE
SUMP
REGULATED OIL PRESSURE
ORIFICE
PRESSURE SIGNAL TO HYDRAULIC CLUTCH
SUMP
HYDRAULIC CLUTCH CONTROL
REGULATED OIL PRESSURE
FREE-TURBINE SERVO

FREE-TURBINE CONTROL COMPONENTS WITH VARIABLE TURBINE NOZZLES

FIGURE 16

The hydromechanical control (Figure 16) protects the free-turbine element from overspeed by a reset head operated from a signal generated by a hydraulic servo. The signal causes the reset-head piston to retract and repositions the gas generator speeder spring to a lower fuel setting, thus removing energy from the free-turbine unit.

The electronic control schematic is similar to those of Figures 7 and 8, except that an overspeed signal from the free-turbine biases the speed summing junction to control impending overspeed.

For discussions of the fluidic and hybrid controls and the controls trade-off, refer to the applicable sections included in the single-shaft control system study.

2.1.2 TURBINE NOZZLE ACTUATOR

The purpose of controlling the free-turbine nozzle area is to provide better fuel economy and aid vehicle acceleration and braking.

Turbine-nozzle positioning is accomplished by an infinitely variable, hydraulic-powered, piston actuator (Figure 16), which moves in response to throttle position. To illustrate, as the throttle is depressed during acceleration, a greater amount of fuel is added to the gas generator, which is the power source for the free turbine. Simultaneously, the free-turbine nozzles are moved to a more open position, allowing the air to flow more freely through the gas generator turbine. In this way, the gas generator is accelerated and power becomes available to the free turbine, in the shortest possible time. When the throttle is retracted, the free-turbine nozzles return to a more closed position and gas generator speed returns to idle. This condition results in a free-wheeling action in the vehicle. If a braking action is desired, the hydraulic brake system pressure overrides the throttle input to the actuator as the brake pedal is depressed and causes the actuator to move the turbine nozzles to a reverse position. The gas stream is now directed upon the free-turbine in a reverse manner, resulting in additional vehicle braking.

Part-throttle fuel economy is accomplished by setting a free-turbine nozzle area at a value to give a constant, maximum-allowable turbine inlet temperature.

The discussion on actuator control and trade-off parameters for the free turbine are identical to that of the single-shaft engine. Therefore, refer to Paragraph 1.3 in the single-shaft study.

2.1.3 Automatic Starting and Protection Control

Discussions of the automatic starting and protection control schemes for the free turbine and single-shaft engines are identical and have already been presented in Paragraph 1.4.

2.1.4 Transmission Control

Transmission control is accomplished through the drive selector and accelerator pedal. Selection of reverse and drive positions causes the transmission to operate similar to existing automotive designs. However, the mechanization of neutral or park requires special provisions in the free-turbine output shaft, thereby preventing its rotation regardless of external load or gas generator output. In addition to locking the free turbine during park, a locking provision on the transmission output shaft prevents the vehicle from rolling.

During vehicle acceleration, the accelerator pedal is depressed, resulting in greater fuel delivery to the engine, while simultaneously requesting a gear change in the transmission through the throttle-biased transmission speed servo. If the acceleration request is large and vehicle velocity low, the transmission will down-shift and transmit a greater amount of torque to the drive wheels, resulting in more rapid acceleration of the vehicle. As the vehicle moves to a higher speed, the transmission is automatically up-shifted to the next lowest gear ratio by the transmission output-shaft speed governor. Should the free turbine tend toward overspeed during any portion of acceleration, an override signal from the free-turbine speed servo will up-shift the transmission. When cruising speed is reached, the transmission will be automatically shifted into the lowest possible gear ratio, resulting in low engine speed and good fuel economy.

Since the transmission is mechanical in nature, a mechanical control for changing gear ratio is more feasible because of the ease of interfacing. Items such as permanent magnet generators and monopoles were considered for sensing transmission output-shaft speed but were ruled out in favor of the mechanical scheme.

2.1.5 Hydraulic Clutch Control

The clutch control shown in Figure 16 has a servo spool that senses a hydraulic pressure signal generated by the free-turbine speed servo. When the free turbine approaches low-speed operation, the clutch servo spool decreases the pressure to the clutch actuator, causing the clutch to slip. This allows the free turbine to idle at a constant minimum speed while transmitting a small torque to the drive wheels. This mode of operation for the slipping-clutch is analogous to a fluid coupling or a torque converter.

A mechanical device was selected in this application because of its hardware simplicity and ease of interfacing with the hydraulically operated clutch.

2.1.6 Sensors

Refer to Paragraph 1.8 for a discussion of the sensors for the free-turbine application.

2.2 CONTROL SYSTEM COMPONENT CONSTRUCTION

Component construction requirements for the free-turbine engine are identical to those of the single-shaft engine. Therefore, refer to Paragraph 1.10 for a discussion of these considerations.

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Number of Pages

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3. Engine Acceleration Versus Horsepower Rating Characteristics 3
4. Prediction Procedure for Gas Turbine Engine Exhaust Emissions 13
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    2. Combustion Efficiency vs CO Emission Index 5
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    4. Stoichiometric Equivalence Ratios 11

APPENDIX 1

AIR POLLUTION CONTROL OFFICE

ADVANCED AUTOMOTIVE POWER SYSTEMS PROGRAM

"Vehicle Design Goals - Six Passenger Automobile"

(Revision C - May 28, 1971 - 11 Pages)

The design goals presented below are intended to provide:

A common objective for prospective contractors.

Criteria for evaluating proposals and selecting a contractor.

Criteria for evaluating competitive power systems for entering first generation system hardware.

Advisory criteria in such areas as rolling resistance, vehicle air drag etc. are included to assist the contractor.

The derived criteria are based on typical characteristics of the class of passenger automobiles with the largest market volume produced in the U. S. during the model years 1969 and 1970. It is noted that emissions, volume and most weight characteristics presented are maximum values while the performance characteristics are intended as minimum values. Contractors and prospective contractors who take exceptions must justify these exceptions and relate these exceptions to the technical goals presented herein.

1. Vehicle weight without propulsion system - W_o.

W_o is the weight of the vehicle without the propulsion system and includes, but is not limited to: body, frame, glass and trim, suspension, service brakes, seats, upholstery, sound absorbing materials, insulation, wheels (rims and tires), accessory ducting, dashboard instruments and accessory wiring, battery, passenger compartment heating and cooling devices and all other components not included in the propulsion system. It also includes accessories such as, the air conditioner compressor, the power steering pump, and the power brakes actuating device.

W_o is fixed at 2700 lbs.

2. Propulsion system weight - W_p.

W_p includes the energy storage unit (including fuel and containment), power converter (including both functional components and controls) and power transmitting components to the driven wheels. It also includes the exhaust system, pumps, motors, fans and fluids necessary for operation of the propulsion system, and any propulsion system heating or cooling devices.

The maximum allowable propulsion system weight, W_{pm}, is 1600 lbs. However, light weight propulsion systems are highly desired. (Equivalent 1970 propulsion system weight with a spark ignition engine is 1300 lbs.)

3. Vehicle curb weight - W_c

$W_c = W_o + W_p$

The maximum allowable vehicle curb weight, W_{cm}, is 4300 lbs. (2700 + 1600 max. = 4300)

4. Vehicle test weight - W_t.

$W_t = W_c + 300$ lbs. W_t is the vehicle weight at which all accelerative maneuvers, fuel economy and emissions are to be calculated. (Items 8c, 8D, 8e).

The maximum allowable test weight, W_{tm}, is 4600 lbs. (2700 + 1600 max. + 300 = 4600).

5. Gross vehicle weight - W_g

$W_g = W_c + 1000$ lbs. W_g is the gross vehicle weight at which sustained cruise grade velocity capability is to be calculated. (Item 8f). The 1000 lbs. load simulates a full load of passengers and baggage.

The maximum allowable gross vehicle weight, W_{gm}, is 5300 lbs. (2700 + 1600 max. + 1000 = 5300).

6. Propulsion system volume - V_p

V_p includes all items identified under item 2. V_p shall be packagable in such a way that the volume encroachment on either the passenger or luggage compartment is not significantly different than today's (1970) standard full size family car. The propulsion system shall not violate the vehicle ground clearance lines as established by the manufacturer of the vehicle used for propulsion system/vehicle packaging. Additionally, the propulsion system shall not violate the space allocated for wheel jounce motions and vehicle steering. Necessary external appearance (styling) changes will be minor in nature. V_p shall also be packagable in such a way that the handling characteristics of the vehicle do not depart significantly from a 1970 full size family car.

The maximum allowable volume assignable to the propulsion system, V_{pm}, is 35 ft.^{3}.

7. Emission Goals

The vehicle when tested for emissions in accordance with the procedure outlined in the November 10, 1970 Federal Register shall have a weight of W_t. The emission goals for the vehicle are:

Hydrocarbons*	- 0.14 grams/mile maximum
Carbon monoxide	- 4.7 grams/mile maximum
Oxides of nitrogen**	- 0.4 grams/mile maximum
Particulates	- 0.03 grams/mile maximum

*Total hydrocarbons (using 1972 measurement procedures) plus total oxygenates. Total oxygenates including aldehydes will not be more than 10 percent by weight of the hydrocarbons or 0.014 grams/mile, whichever is greater.

**measured or computed as NO_2.

8. Start up, Acceleration, and Grade Velocity Performance.

a. Start up:

The vehicle must be capable of being tested in accordance with the procedure outlined in the November 10, 1970 Federal Register without special driver startup/warmup procedures.

The maximum time from key on to reach 65 percent full power is 45 sec. Ambient conditions are 14.7 psia pressure, 60°F temperature.

Powerplant starting techniques in low ambient temperatures shall be equivalent to or better than the typical automobile spark-ignition engine. Conventional spark-ignition engines are deemed satisfactory if after a 24 hour soak at -20°F the engine achieves a self-sustaining idle condition without further driver input within 25 seconds. No starting aids external to the normal vehicle system shall be needed for -20°F starts or higher temperatures.

b. Idle operation conditions:

The fuel consumption rate at idle operating condition will not exceed 14 percent of the fuel consumption rate at the maximum design power condition. Recharging of energy storage systems is exempted from this requirement. Air conditioning is off, the power steering pump and power brake actuating device, if directly engine driven, are being driven but are unloaded.

The torque at transmission output during idle operation (idle creep torque) shall not exceed 40 foot-pounds, assuming conventional rear axle ratios and tire sizes. This idle creep torque should result in level road operation in high gear which does not exceed 18 mph.

c. Acceleration from a standing start:

The minimum distance to be covered in 10.0 sec. is 440 ft. The maximum time to reach a velocity of 60 mph is 13.5 sec. Ambient conditions are 14.7 psia, 85° F. Vehicle weight is W_t. Acceleration is on a level grade and initiated with the engine at the normal idle condition.

d. Acceleration in merging traffic:

The maximum time to accelerate from a constant velocity of 25 mph to a velocity of 70 mph is 15.0 sec. Time starts when the throttle is depressed. Ambient conditions are 14.7 psia, 85° F. Vehicle weight is W_t, and acceleration is on level grade.

e. Acceleration, DOT High Speed Pass Maneuver:

The maximum time and maximum distance to go from an initial velocity of 50 mph with the front of the automobile (18 foot length assumed) 100 feet behind the back of a 55 foot truck traveling at a constant 50 mph to a position where the back of the automobile is 100 feet in front of the front of the 55 foot truck is, 15 sec. and 1400 ft. The entire maneuver takes place in a traffic lane adjacent to the lane in which the truck is operated. Vehicle will be accelerated until the maneuver is completed or until a maximum speed of 80 mph is attained, whichever occurs first. Vehicle acceleration ceases when a speed of 80 mph is attained, the maneuver then being completed at a constant 80 mph. (This does not imply a design requirement limiting the maximum vehicle speed to 80 mph.) Time starts when the throttle is depressed. Ambient conditions are 14.7 psia, 85° F. Vehicle weight is W_t, and acceleration is on level grade.

f. Grade velocity:

The vehicle must be capable of starting from rest on a 30 percent grade and accelerating to 15 mph and sustaining it. This is the steepest grade on which the vehicle is required to operate in either the forward or reverse direction.

The minimum cruise velocity that can be continuously maintained on a 5 percent grade with an accessory load of 4 hp shall be not less than 60 mph.

The vehicle must be capable of achieving a velocity of 65 mph up a 5 percent grade and maintaining this velocity for a period of 180 seconds when preceded and followed by continuous operation at 60 mph on the same grade (as above).

The vehicle must be capable of achieving a velocity of 70 mph up a 5 percent grade and maintaining this velocity for a period of 100 seconds when preceded and followed by continuous operation at 60 mph on the same grade (as above).

The minimum cruise velocity that can be continuously maintained on a level road (zero grade) with an accessory load of 4 hp shall be not less than 85 mph with a vehicle weight of W_t.

Ambient conditions for all grade specifications are 14.7 psia 85° F. Vehicle weight is W_g for all grade specifications except the zero grade specification.

The vehicle must be capable of providing performance (Paragraphs 8c, 8d, 8e 8f)within 5 percent of the stated 85° F values, when operated at ambient temperatures from -20° F to 105° F.

9. Minimum vehicle range:

Minimum vehicle range without supplementing the energy storage will be 200 miles. The minimum range shall be calculated for, and applied to each of the two following modes: 1) A city-suburban mode, and 2) a cruise mode.

Mode 1: Is the driving cycle which appears in the November 10, 1970 Federal Register. For vehicles whose performance does not depend on the state of energy storage, the range may be calculated for one cycle and ratioed to 200 miles. For vehicles whose performance does depend on the state of energy storage the Federal driving cycle must be repeated until 200 miles have been completed.

Mode 2: Is a constant 70 mph cruise on a level road for 200 miles.

The vehicle weight for both modes shall be, initially, W_t. The ambient conditions shall be a pressure of 14.7 psia, and temperatures of 60° F, 85° F and 105° F. The vehicle minimum range shall not decrease by more than 5 percent at an ambient temperature of -20° F.

For hybrid vehicles, the energy level in the power augmenting device at the completion of operation will be equivalent to the energy level at the beginning of operation.

10. System thermal efficiency:

System thermal efficiency will be calculated by two methods:

A. A "fuel economy" figure based on 1) miles per gallon (fuel type being specified) and 2) the number of Btu per mile required to drive the vehicle over the 1972 Federal driving cycle which appears in the November 10, 1970 Federal Register. Fuel economy is based on the fuel or other forms of energy delivered at the vehicle. Vehicle weight is W_t.

B. A "fuel economy" figure based on 1) miles per gallon (fuel type being specified) and 2) the number of Btu per mile required to drive the vehicle at constant speed, in still air, on level road, at speeds of 20, 30, 40, 50, 60, 70, and 80 mph. Fuel economy is based on the fuel or other forms of energy delivered at the vehicle. Vehicle weight is W_t.

In both cases, the system thermal efficiency shall be calculated with sufficient electrical, power steering and power brake loads in service to permit safe operation of the automobile. Calculations shall be made with and without air conditioning operating. The ambient conditions are 14.7 psia and temperatures of 60° F, 85° F and 105° F. Calculations shall be made with heater operating at ambient conditions of 14.7 psia and 30° F (18,000 Btu/hr).

11. Air Drag Calculation:

The product of the drag coefficient, C_d, and the frontal area, A_f, is to be used in air drag calculations. The product C_dA_f has a value of 12 ft^2. The air density used in computations shall correspond to the applicable ambient air temperature.

12. Rolling Resistance:

Rolling resistance, R, is expressed in the equation $R = W/65\ [1 + (1.4 \times 10^{-3}V) + (1.2\ 10^{-5}V^2)]$ lbs. V is the vehicle velocity in ft/sec. W is the vehicle weight in lbs.

13. Accessory power requirements:

The accessories are defined as subsystems for driver assistance and passenger convenience, not essential to sustaining the engine operation and include: the air conditioning compressor, the power steering pump, the alternator (except where required to sustain operation), and the power brakes actuating device. The accessories also include a device for heating the passenger compartment if the heating demand is not supplied by waste heat.

Auxiliaries are defined as those subsystems necessary for the sustained operation of the engine, and include condensor fan(s), combustor fan(s), fuel pumps, lube pumps, cooling fluid pumps, working fluid pumps and the alternator when necessary for driving electric motor driven fans or pumps.

The maximum intermittent accessory load, P_{aim}, is 10 hp (plus the heating load, if applicable). The maximum continuous accessory load, P_{acm}, is 7.5 hp (plus the heating load if applicable). The average accessory load, P_{aa}, is 4 hp.

If accessories are driven at variable speeds, the above values apply. If the accessories are driven at constant speed, P_{aim} and P_{acm} will be reduced by 3 hp.

14. Passenger comfort requirements:

Heating and air conditioning of the passenger compartment shall be at a rate equivalent to that provided in the present (1970) standard full size family car.

Present practice for maximum passenger compartment heating rate is approximately 30,000 Btu/hr. For an air conditioning system at 110° F ambient, 80° F and 40% relative humidity air to the evaporator, the rate is approximately 13,000 Btu/hr.

15. Propulsion system operating temperature range:

The propulsion system shall be operable within an expected ambient temperature range of -40° to 125° F.

16. Operational life:

The mean operational life of the propulsion system should be approximately equal to that of the present spark-ignition engine. The mean operational life should be based on a mean vehicle life of 105,000 miles or ten years, whichever comes first.

The design lifetime of the propulsion system in normal operation will be 3500 hours. Normal maintenance may include replacement of accessable minor parts of the propulsion system via a usual maintenance procedure, but the major parts of the system shall be designed for a 3500 hour minimum operation life.

The operational life of an engine shall be determined by structural or functional failure causing repair and replacement costs exceeding the cost of a new or rebuilt engine. (Functional failure is defined as power degradation exceeding 25 percent or top vehicle speed degradation exceeding 9 percent).

17. Noise standards: (Air conditioner not operating)

a. Maximum noise test:

The maximum noise generated by the vehicle shall not exceed 77 dbA when measured in accordance with SAE J986a. Note that the noise level is 77 dbA whereas in the SAE J986a the level is 86 dbA.

b. Low speed noise test:

The maximum noise generated by the vehicle shall not exceed 63 dbA when measured in accordance with SAE J986a except that a constant vehicle velocity of 30 mph is used on the pass-by, the vehicle being in high gear or the highest gear in which it can be operated at that speed.

c. Idle noise test:

The maximum noise generated by the vehicle shall not exceed 62 dbA when measured in accordance with SAE J986a except that the engine is idling (clutch disengaged or in neutral gear) and the vehicle passes by at a speed of less than 10 mph. the microphone will be placed at 10 feet from the centerline of the vehicle pass line.

18. Safety standards:

The vehicle shall comply with all current Department of Transportation Federal Motor Vehicle Safety Standards. Reference DOT/HS 820 083.

19. Reliability and maintainability:

The reliability and maintainability of the vehicle shall equal or exceed that of the spark-ignition automobile. The mean-time-between failure should be maximized to reduce the number of unscheduled service trips. All failure modes should not represent a serious safety hazard during vehicle operation and servicing. Failure propagation should be minimized. The power plant should be designed for ease of maintenance and repairs to minimize costs, maintenance personnel education, and downtime. Parts requiring frequent servicing shall be easily accessable.

20. Cost of ownership:

The net cost of ownership of the vehicle shall be minimized for ten years and 105,000 miles of operation. The net cost of ownership includes initial purchase price (less scrap value), other fixed costs, operating and maintenance costs. A target goal should be to not exceed 110 percent of the average net cost of ownership of the present standard size automobile with spark-ignition engine as determied by the U.S. Department of Commerce 1969-70 statistics on such ownership.

TABLE 1

OFFICE OF AIR PROGRAMS SIMPLIFIED DRIVING CYCLE

Route	Average Speed, mph	Time, hr	Percent of Time
Federal Driving Cycle	19.84	1750	50
Simplified Suburban Route (equal times at Constant 20, 30, and 40 mph)	30.00	1150	33
Simplified Country Route (equal times at Constant 50, 60, and 70 mph)	60.00	600	17
TOTALS	30	3500	100

APPENDIX 2

DE-IONIZED WATER SUPPLY REQUIREMENTS FOR GAS TURBINE AUTOMOBILE DRIVE

Purified water may be used to increase the transient power capability of the gas turbine engine. Thus, in the automobile drive, an engine of smaller maximum airflow may reduce the power plant installation size and weight as well as cost. A water flow of 6 percent of the airflow by mass gives 30 percent more power with the same turbine inlet temperature (TIT) for higher ambient temperatures. As a result, a single-shaft gas turbine can, on a 105°F day, deliver 110 hp instead of 85, with approximately 1.0 lb/sec air, using 1900°F TIT and a fuel-to-air ratio of 0.017. Thus, the water flow is about 0.06/0.017 = 3.5 times the fuel flow.

The portion of time involved in a maximum power maneuver for a typical commuter duty cycle is estimated as follows:

Total trip time = 20 min
Average speed = 45 mph
Trip distance = 15 mi

Operation	Duration, min	Number
Maximum acceleration at stop	0.50	2
Maximum acceleration - merging traffic	0.25	1
Maximum acceleration - high-speed pass	0.25	1
TOTAL time, min	1.00	

This would require a water quantity of 3.6 lb or 0.43 gal. If the water tank were filled on the same basis as the fuel tank, the water tank size would be dependent on the vehicle miles per gallon of fuel:

Fuel Tank	Fuel, mpg	Fuel, gal/trip	Water, gal/trip	Water Tank, gal
20 gal	10	1.50	0.43	5.75
	13	1.15	0.43	7.47
	15	1.00	0.43	8.60
	20	0.75	0.43	11.47
	25	0.60	0.43	14.30

From the preceding, it would be desirable to supply the vehicle with about a 10-gal tank. The tank could be blow-molded polyethylene and could be pressurized by compressor bleed-air regulated to about 15 psig maximum pressure. A water-quantity indicator on the dash would be needed to assure the operator of the availability of the water for the critical power needed at various times.

The quantity of purified water required for vehicle operation is sufficient to warrant consideration of a means of supply for motorists. While de-ionized water can be produced in large quantities (from 1000 to 2000 gph) at costs of \$0.20 to \$0.25/1000 gal, shipping, storage, and handling costs would force the consumer price to about \$0.03/gal, determined as follows:

Distribution, 2000 gal, \$30.00/trip = \$0.015/gal

Storage, pumps, amortization = \$0.015/gal

Handling at service station, 5 gal, 1 min, \$2.00/hr = \$0.003/gal

TOTAL = \$0.033/gal

An alternate approach would be a reverse-osmosis and pump system to produce purified water of 20- to 30-ppm purity from city tap water of 400 to 600 ppm. In this instance, a boost pump supplying 600 psig of water from the city tap to a module for 700 gal/day would cost

Fuel Tank	Fuel, mpg	Fuel, gal/trip	Water, gal/trip	Water Tank, gal
20 gal	10	1.50	0.43	5.75
	13	1.15	0.43	7.47
	15	1.00	0.43	8.60
	20	0.75	0.43	11.47
	25	0.60	0.43	14.30

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Storage, pumps, amortization	= $0.015/gal
Handling at service station, 5 gal, 1 min, $2.00/hr	= $0.003/gal
TOTAL	= $0.033/gal

An alternate approach would be a reverse-osmosis and pump system to produce purified water of 20- to 30-ppm purity from city tap water of 400 to 600 ppm. In this instance, a boost pump supplying 600 psig of water from the city tap to a module for 700 gal/day would cost approximately \$1000.00. A service station pumping 10,000 gal of gasoline per day would require 4300 gal of water (at an average fuel consumption of 15 mpg), necessitating six reverse-osmosis module systems at a cost of \$6000.00. Assuming replacement every two years, total production would be $2(360)4300 = 3.10 \times 10^6$ gal. Thus, water production cost would be $6000(1.0)/3.10 \times 10^6$ gal. Thus, water production cost would be $6000(1.0)/3.10 \times 10^6 = \$0.00194$/gal. Other costs include a minor storage tank and service-handling costs at \$0.0037/gal or a total of \$0.0060/gal. These costs are summarized as follows:

	Cents/gal
Amortizing reverse-osmosis system	0.19
Electricity	0.05
Storage tank	0.03
Handling	0.33
	0.60
Profit	0.40
Sales price	1.00

This would increase the fuel cost to the consumer, dependent on the vehicle miles per gal, as follows:

Fuel, mpg	Additional Cost of Water, cents/gal fuel
10	0.29
13	0.37
15	0.43
20	0.57
25	0.71

APPENDIX 3

ENGINE ACCELERATION VERSUS HORSEPOWER RATING CHARACTERISTICS

For a given engine configuration, it is desirable to express the relationship between the time required to accelerate between two values of percent speed and the engine peak-power level, when the configuration is scaled geometrically.

Denote the horsepower level at maximum turbine inlet temperature by P. The value, P, is then a single-valued function of the shaft speed, N (engine speed for a single-shaft engine or gas generator speed for a free-turbine engine), for this turbine inlet temperature. Also, let:

W = engine throughflow rate

D = compressor or turbine tip diameter

I = engine polar mass moment of inertia

T = engine output torque

t = time

$n = N/N_{max}$ = percent shaft speed

Also, let superscripts [1] and [2] refer to the original values of the engine variables listed above and the values for the scaled engine, respectively.

Holding all other variables constant, the following relationships are applicable:

$$P \propto^{*} W$$

$$W \propto D^2$$

$$I \propto D^5, \text{ for a geometrical scale} \tag{1}$$

$$T \propto \frac{P}{N}$$

Also,

$$T = kI \frac{dN}{dt} = kI\, N_{max} \frac{dn}{dt}, \; k = \text{constant} \tag{2}$$

and,

$$DN = k_1, \text{ a constant} \tag{3}$$

for a given configuration. Therefore, from Eq. (1) and (3),

$$P \propto \frac{1}{N^2} \tag{4}$$

Then, using Eq. (2),

$$\Delta t^{[1]} = kN_{max}^{[1]} \int_{n_1}^{n_2} \frac{I^{[1]}}{T^{[1]}}\, dn \tag{5}$$

Also,

$$\Delta t^{[2]} = kN_{max}^{[2]} \int_{n_1}^{n_2} \frac{I^{[2]}}{T^{[2]}}\, dn \tag{6}$$

But from Eq. (1),

$$N_{max}^{[2]} = \left(\frac{P^{[1]}}{P^{[2]}}\right)^{1/2} \left(N_{max}^{[1]}\right) \tag{7}$$

---

*$\propto$ denotes proportionality, e.g., $A \propto B$ means A is proportional to B.

$$I^{[2]} = \left(\frac{P^{[2]}}{P^{[1]}}\right)^{5/2} (I^{[1]})$$

$$T^{[2]} = \left(\frac{P^{[2]}}{P^{[1]}}\right)^{3/2} (T^{[1]})$$

Substituting Eq. (7) into Eq. (6) yields

$$\Delta t^{[2]} = k\left(\frac{P^{[2]}}{P^{[1]}}\right)^{1/2} N_{max}^{[1]} \int_{n_1}^{n_2} \frac{I^{[1]}}{T^{[1]}}\, dn$$

or simplifying

$$\Delta t^{[2]} = \left(\frac{P^{[2]}}{P^{[1]}}\right)^{1/2} \Delta t^{[1]} \qquad (8)$$

Therefore, when the engine is scaled to a higher power level, more time is required to accelerate the engine (for a single-shaft engine or gas generator for a free-turbine engine) from 1-percent speed, such as idle, to another percent speed, such as maximum engine speed.

APPENDIX 4

PREDICTION PROCEDURE FOR GAS TURBINE ENGINE EXHAUST EMISSIONS

1. CARBON MONOXIDE (CO) AND TOTAL HYDROCARBONS (THC)

Prediction of CO and THC has been obtained by empirical correlation of the weight ratio of THC to CO as a function of combustion efficiency. Figure 1 shows that this relationship is approximately linear with combustion efficiency and is similar for similar combustor types. In general, the annular combustors, with higher wall surface areas, produce a greater ratio of THC to CO compared with can-type combustors. The ratio can be estimated by the following equation:

$$R = R_{99} \, (110 - \eta_c)$$

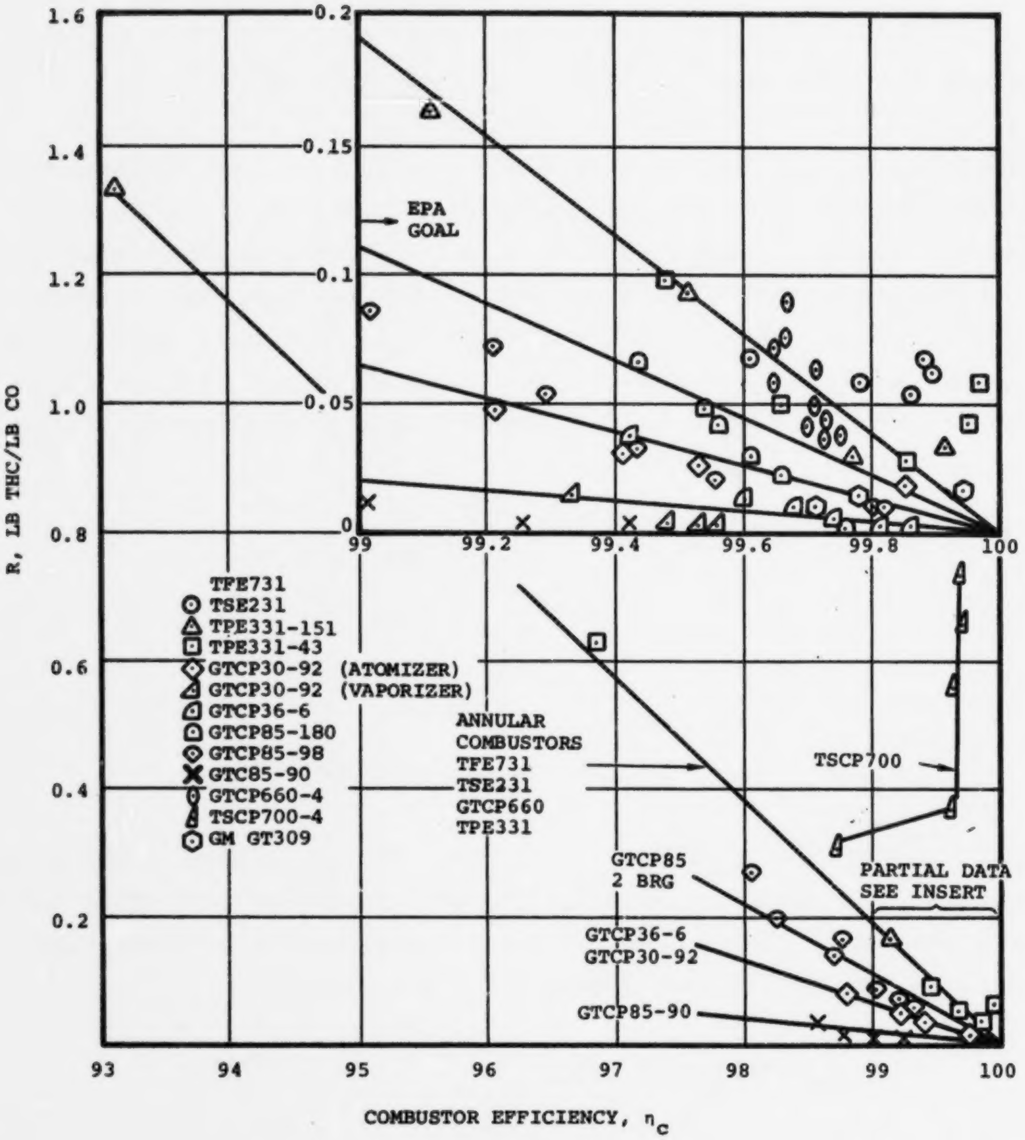
where:

R = ratio, lb THC/lb CO

R_{99} = R at 99 percent combustion efficiency

Values of R_{99} for approximate fit to the data are listed. These values were selected with emphasis on low efficiency points where emissions are more critical. Linearity is less apparent at very high efficiencies, as seen in the insert of Figure 1.

Combustor Type	R_{99}
Annular (731,231,660,331)	0.19
6-in. can (2 brg 85)	0.11
5-in. can (30-92, 36-6)	0.065
5-in. can (4 brg 85)	0.02



RATIO LB THC/LB CO VS COMBUSTION EFFICIENCY

FIGURE 1

The EPA automotive goals of 0.41 gm/mi THC and 3.4 gm/mi CO give a value for R of 0.1205. One engine, the TSCP700, shows an entirely different characteristic on Figure 1 from the other annular combustors. This is the result of a constant level of measured THC (expressed as lb THC per 1000 lb fuel) over the load range. There is some indication that the regenerated engines may not have a linear relation but a constant value of R on the order of 0.03. Further data analysis is required to resolve this.

From a given value of R, calculation of the emission levels requires knowledge of the combustion efficiency; maps are available for most combustors. Combustion efficiencies from emission data are calculated directly from measured data by an equation derived from the ratio of actual-to-ideal chemical enthalpy release:

$$\eta_c = 100 - \frac{E_{CO}\ (HV_{CO} + R \times HV_{THC})}{10 \times LHV_f}$$

where:

η_c = combustion efficiency, percent

E_{CO} = CO emission index, lb CO per 1000-lb fuel

HV_{CO} = heating value of CO = 4345 Btu/lb

$HV_{THC} = LHV_f + H_v$

LHV_f = fuel lower heating value, Btu/lb

H_v = heat of vaporization = 156 Btu/lb

For typical fuels, LHV_f is 18,500 and

$$\eta_c = 100 - E_{CO} (0.0235 + R \times 0.1008)$$

If η_c and R are given, E_{CO} can be calculated and the emission index of THC, E_{THC}, can be found from

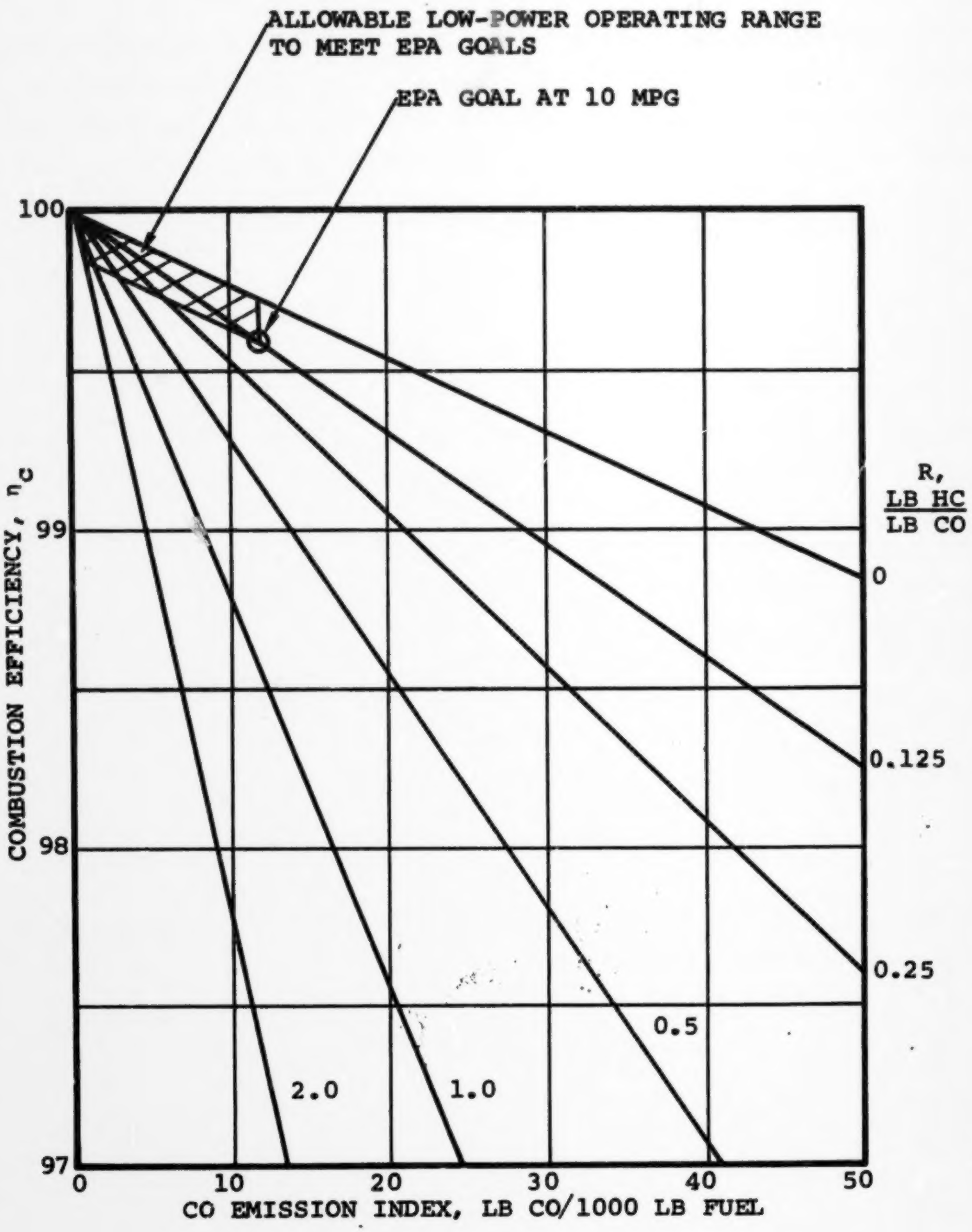
$$E_{THC} = R \times E_{CO}$$

Figure 2 displays the relationship of combustion efficiency as a function of CO emission index for various THC-CO weight ratios. Achievement of the EPA goals requires combustion efficiencies greater than 99.5 percent, including critical-idle and low-power conditions.

Major assumptions in this expression for efficiency are that all hydrocarbons be in the form of original fuel and no free hydrogen be present. EPA specifications express THC as $CH_{1.85}$, which is representative of most fuels. Measured emission data are variously expressed in terms of carbon, hexane, or methane, and care must be taken that proper weight conversions are employed in data reduction. In actual fact, many hydrocarbon compounds are present, each with a different heating value. Present practical measurement equipment cannot identify the separate compounds. Therefore, the above expression represents the best estimate of combustion efficiency believed to be more accurate than conventional temperature-weight-flow techniques.

2. OXIDES OF NITROGEN

Oxides of nitrogen are not products of inefficient combustion and cannot be related directly to combustion efficiency. Nitric oxide, NO, is the product of reaction of atmospheric nitrogen and oxygen that occurs at temperatures in excess of 3000°F. Residence time within the



COMBUSTION EFFICIENCY VS CO EMISSION INDEX

FIGURE 2

flame zone, fortunately, is not sufficient for this reaction to reach equilibrium. However, this complicates the analysis, by requiring consideration of the rate kinetics of the specific formation processes. Extensive research has proven that two reactions are primarily responsible for NO formation:

$$N_2 + O \rightleftarrows NO + N \qquad (1)$$

$$N + O_2 \rightleftarrows NO + O \qquad (2)$$

The net formation rate of NO can be written as

$$\frac{d(NO)}{dt} = k_{1f}\,(N_2)(O) + k_{2f}\,(N)(O_2) - k_{1b}\,(NO)(N) - k_{2b}\,(NO)(O)$$

where:

k = kinetic rate constants for reactions 1 and 2

f = forward

b = backward

(x) = concentration of species N_2, O_2, NO, N

If all species concentrations except NO are assumed constant, the rate equation can be integrated to

$$\frac{(NO)}{(NO)_e} = 1 - e^{-t/\tau NO}$$

where:

$(NO)_e$ = NO concentration at equilibrium

t = elapsed residence time, msec

τ_{NO} = reaction time constant, msec

$(NO)_e$ can be found either from an equilibrium composition for the mixture or by setting the rate equation to zero and solving for (NO). The reaction time constant, τ_{NO}, is

$$\tau_{NO} = \frac{1000}{k_{1b}\ (N) + k_{2b}\ (O)}$$

Rate constants are functions of mixture temperature only. Fuel-air ratios at or near stoichiometric produce the highest flame temperatures with resultant, substantially greater, NO formation rates.

A model by Heywood* integrates the formation rate for statistical distributions of mixture ratio and residence time. This model can be simplified by assuming that all NO is formed in an equivalent stoichiometric mixture. If the flow is distributed uniformly along the length of the combustor, residence time of the initial mixture at the dome is approximately

$$t = \frac{V_c \sigma_s \times 1000}{W_c}$$

where:

t = residence time, msec
V_c = combustor volume, ft^3
σ_s = density of stoichiometric mixture
W_c = combustor airflow, lb/sec

The reaction temperature is obtained from an adiabatic enthalpy balance for a stoichiometric mixture with dissociation. Effects of inlet relative humidity on flame temperature are included. The equilibrium procedure of NASA TN-D 4747 is employed, which provides species mole fractions that would be attained with NO formation suppressed. The temperature and NO-free composition are then used to compute a formation rate and subsequent NO level for the specified residence time.

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*Heywood, J. B., "Gas Turbine Combustor Modeling for Calculating Nitric Oxide Emissions," AIAA Paper No. 71-712, June 1971.

The NO_x concentration, for a stoichiometric mixture resulting from the foregoing calculation, can be converted to an emission index expressed in pounds of NO_x/1000 lb of fuel by the following expression:

$$EI_s = 1000 \times (NO)_s \times \frac{M_{NO2}}{M_s} \times \frac{1}{f_s}$$

where:

EI_s = Emission index of stoichiometric mixture, pounds NO_2/1000 lb of fuel

$(NO)_s$ = Mole fraction of NO formed in a stoichiometric mixture

M_{NO_2} = Molecular weight of NO_2

M_s = Molecular weight of stoichiometric mixture

f_s = Fuel-air ratio of stoichiometric mixture

Three procedures were investigated for correlating the calculated stoichiometric emission index with the measured NO_x emission index levels for a number of the engines tested during the EPA test program. The first procedure consisted of adjusting the combustor volume to attempt to find a constant characteristic fraction of the total combustor volume for each engine that would correlate the data. This was successfull for only the TFE731 and TEP331 engines. For these two the characteristic volume fraction was of the order of 20 percent of the total combustor volume.

The second procedure was a modification of Heywood's model. Heywood assumed a statistical distribution for both residence time and fuel-air ratio in the primary zone. Integration of the formation rates over these distributions then yields an overall emission index. The AiResearch modification considered of assuming that all of the NO forms in a stoichiometric mixture. The amount of mixture was computed as a fraction of primary zone overall fuel-air ratio, using the distribution relation given by Heywood. A single residence time was assumed. A standard deviation for the distribution function was then computed, with which the computed and measured emissions would agree. This method failed to give good agreement over the range of engine fuel-air ratios. The standard deviation required at low-power conditions was considerably greater than for full power.

The third procedure proved to be most satisfactory. This was simply to express the measured emission index as a fraction of the computed stoichiometric emission index. With a few exceptions, this fraction remains relatively constant for a given engine over the load range. Figure 3 is a plot of this effective fraction of stoichiometric mixture emission index for a number of the engines evaluated in the EPA test program. The combustor inlet temperatures were not recorded in the test program and were estimated from measured compressor exit pressure with the assumption of 80-percent compressor efficiency.

The value of the fraction at stoichiometric equivalence ratio exhibits a general trend with design-point combustor loading, i.e., more highly loaded combustors have a higher effective fraction. This relationship (Figure 4) implies that, as combustor size is reduced, a larger fraction of the primary-zone volume is involved in the fuel-burning process.

The prediction procedure can then be summarized. Combustor design-point loading is calculated from

$F_e = EI_M / EI_S$

EI_M = MEASURED NO EMISSION INDEX, $\frac{\text{LB } NO_2}{\text{1000 LB FUEL}}$

EI_S = COMPUTED EMISSION INDEX FOR A STOICHIOMETRIC MIXTURE, $\frac{\text{LB } NO_2}{\text{1000 LB FUEL}}$

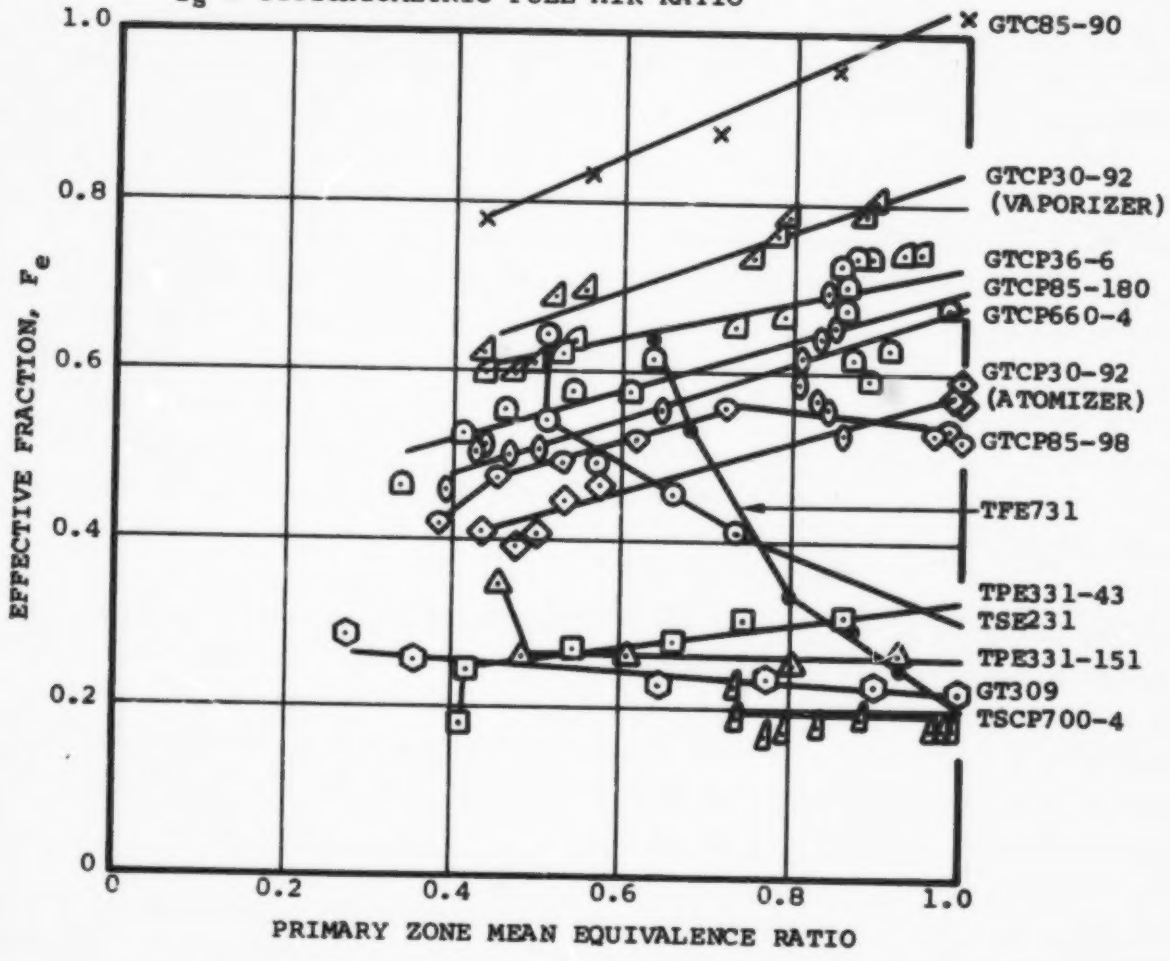
$$= \frac{1000 \quad (NO)_S \quad M_{NO_2}}{M_S \quad f_s}$$

$(NO)_S$ = MOLE FRACTION OF NO FORMED AT STOICHIOMETRIC CONDITIONS DURING CHARACTERISTIC COMBUSTOR RESIDENCE TIME

M_{NO_2} = MOLECULAR WEIGHT OF NO_2

M_S = MOLECULAR WEIGHT OF STOICHIOMETRIC MIXTURE

f_s = STOICHIOMETRIC FUEL AIR RATIO



EFFECTIVE FRACTION OF STOICHIOMETRIC AIR FOR MEASURED NO_X

FIGURE 3

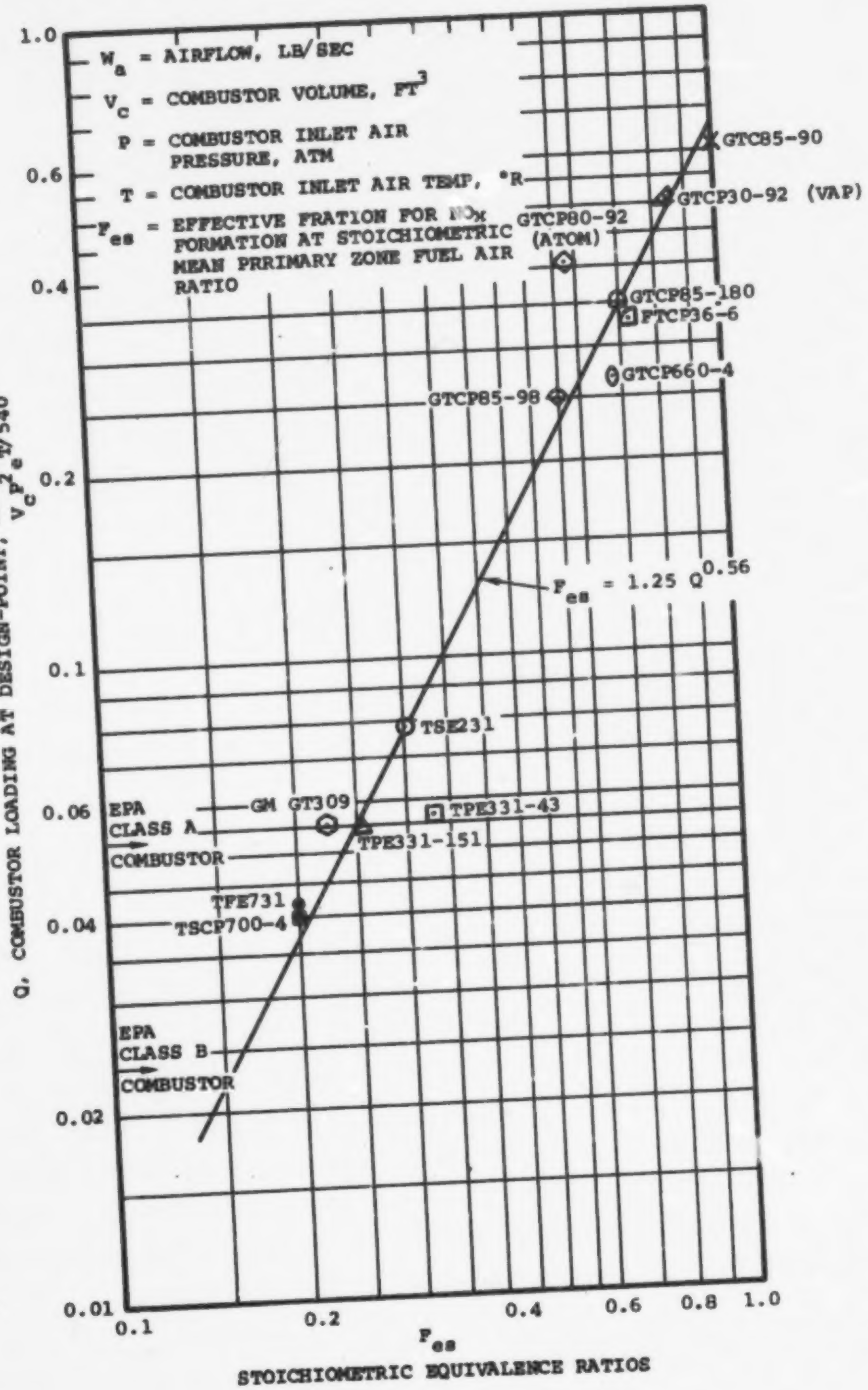

W_a = AIRFLOW, LB/SEC
V_c = COMBUSTOR VOLUME, FT^3
P = COMBUSTOR INLET AIR PRESSURE, ATM
T = COMBUSTOR INLET AIR TEMP, °R
F_{es} = EFFECTIVE FRATION FOR NO_x FORMATION AT STOICHIOMETRIC MEAN PRRIMARY ZONE FUEL AIR RATIO
GTC85-90
GTCP30-92 (VAP)
GTCP80-92 (ATOM)
GTCP85-180
FTCP36-6
GTCP660-4
GTCP85-98
$F_{es} = 1.25\ Q^{0.56}$
TSE231
GM GT309
TPE331-43
TPE331-151
EPA CLASS A COMBUSTOR
TPE731
TSCP700-4
EPA CLASS B COMBUSTOR
1.0
0.6
0.4
0.2
0.1
0.06
0.04
0.02
0.01
0.1
0.2
0.4
0.6
0.8
1.0
Q, COMBUSTOR LOADING AT DESIGN-POINT, $\frac{W_a}{V_c P^2 e^{T/540}}$
F_{es}
STOICHIOMETRIC EQUIVALENCE RATIOS

FIGURE 4

$$Q_d = \frac{W_a}{P^2 V_c e^{T/540}}$$

where:

Q_d = combustor loading at design-point

W_a = air mass flow rate, lb/sec

P = inlet air pressure, atm

V_c = combustor volume, ft^3

T = inlet air temperature, °R

This loading parameter better correlated the data, compared with others commonly employed. The effective fraction at stoichiometric equivalence ratio, F_{es}, is calculated from

$$F_{es} = 1.25\ Q_d^{0.56}$$

The fraction can be assumed constant, or a linear relationship can be used if data for a similar combustor is available as from Figure 3. With this fraction and the stoichiometric NO concentration calculated from the residence time, the emission index of pounds NO/1000 lb of fuel can be calculated. This is then converted to NO_2, to be consistent with specified limits. No method has yet been devised for characterizing the relative ratio of NO to NO_2. Presumably, exhaust NO_2 is formed in the engine exhaust system or perhaps the emission sampling line by oxidation of the NO formed in the combustor.

For the EPA cycle optimization program, the cycles being evaluated are regenerated, and the low-emission combustor for the EPA Combustor Program will be used to establish combustor performance criteria for relative evaluation of emissions from the various cycles. The estimated combustion efficiency map for this combustor is being employed for every cycle. Resizing the combustor for each cycle will be considered, if the low-combustion combustor proves to be far from optimum. From the design-point loading, Figure 4 gives an equivalent stoichiometric fraction of between 0.2 and 0.25. Accordingly, a value of 0.2 will be used and assumed constant for all engine conditions.

For CO and THC estimates, the EPA goal of 0.125 will be used at 99-percent efficiency. At actual operating efficiencies for the cycles, this will produce THC-CO ratios consistent with the General Motors regenerated GT 309. Recent data on AiResearch regenerated engines will be used to refine this value.

APPENDIX 5

MANUFACTURING COST ESTIMATE FOR FULL-SIZE, SIX-PASSENGER AUTOMOBILE, V-8 ENGINE

According to marketing and financial managers in the automobile industry, the markup in price from factory manufacturing cost to consumer list price is divided in approximately the following manner:

Factory cost x 1.20 = Distributor cost
Distributor cost x 1.20 = Dealer cost
Dealer cost x 1.20 = Consumer list price

In addition, manufacturing costs of a full-size automobile are divided among several groups of components in approximately the following proportions:

(a) Cost of body, interior, wheels and tires = 50 percent of total manufacturing cost

(b) Cost of the engine = 20 percent of total cost

(c) Cost of the transmissions = 15 percent of total cost

(d) Cost of the differential, driveline, and axles = 15 percent of total cost

Consumer cost price is obtained from list price according to the following equation*:

Consumer cost = List price + sales tax + shipping cost + advertising cost

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* "Buyer's Guide Magazine", Publications International, LTD, 1970

Consumer cost is assumed to be $3374.00 (refer to Appendix 5) and that:

Sales tax = 4 percent of list price

Advertising cost = $20.00

Shipping cost = $156.00

Therefore, list price = $3075.00, and factory cost is $1780.00.

Using the preceding distribution of costs, the component costs shown in Table 1 are determined for a list price of $3075.00.

TABLE 1

COMPONENT COST

Item	Manufacturing Cost, Dollars
Body, interior, wheels, and tires	890.00
Engine, including controls and engine-mounted accessories	356.00
Transmission	267.00
Driveline and axles	267.00

APPENDIX 6

ESTIMATED COST OF OPERATING AN AUTOMOBILE: 1970*

The following estimates were based on a four-door sedan, costing $3374.00, including $189.00 Federal excise tax, with an assumed life-span of 10 years, 100,000 miles:

Item	10-Year Total Cost	Cents Per Mile Cost, 1970 (January)					
	1970 (Jan.)	10-year average	1st year (11,500 miles)	3d year (11,500 miles)	5th year (9,900 miles)	7th year (9,500 miles)	10th year (5,700 miles)
Total	$11,890	11.89	11.21	12.10	11.50	12.02	10.82
Costs excluding taxes	10,537	10.54	11.93	10.88	10.32	10.88	9.56
Depreciation	3,185	3.19	6.59	3.92	2.60	1.63	0.88
Repairs and Maintenance (a)	1,521	1.52	0.50	1.59	1.74	3.40	0.53
Replacement tires and tubes (b)	385	0.39	0.12	0.12	0.39	0.44	0.82
Accessories (c)	28	0.03	0.01	0.01	0.01	0.05	0.06
Gasoline (d)	1,733	1.73	1.73	1.73	1.73	1.73	1.73
Oil (d)	158	0.16	0.11	0.13	0.16	0.19	0.22
Insurance (e)	1,722	1.72	1.41	1.73	1.87	1.57	2.61
Garaging, parking, tolls, etc. (f)	1,805	1.80	1.43	1.65	1.82	1.88	2.71
Taxes and fees (g)	1,353	1.35	2.28	1.22	1.18	1.14	1.26

(a) Includes lubrication, washing, and waxing; replacement of spark plugs, points and condenser, wiper blades, fan belt, radiator hoses, etc.; starter, water pump, and brake overhaul; universal-joint replacement, etc.; and major repair such as a complete valve job.

(b) Covers 7 new regular tires and 4 new snow tires during life of car.

(c) Includes a set of vinyl floor mats and seat covers.

(d) Gasoline use set at 13.8 miles per gallon; oil use associated with gasoline at rate of 1 to 128.

(e) Includes $50,000 combined public liability, property damage, $1,000 medical, and comprehensive for full 10 years, uninsured motorist coverage, and $100 deductible collision insurance assumed for first 5 years.

(f) Includes monthly charges of $10 for garage rental or cost of owner's garaging facility, parking fee average of $54 per year, and toll average of $6.50 per year.

(g) Includes Federal gasoline tax of 4 cents and Maryland gasoline tax of 7 cents per gallon: Maryland registration fee of $20 and titling tax at 4 percent of retail price; and Federal excise taxes on motor vehicles, tires, and oil. Total taxes include property and oil taxes.

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* Statistical Abstracts of the U.S., U.S. Department of Commerce, Bureau of Census, 1971, p.547, No. 846.